HEAT TRANSFER

TO

BOILING LIQUIDS

A THESIS

SUBMITTED BY

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C O N T E N T S

PART III

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ABSTRACT

For the determination of the boiling film coefficients a knowledge of the temperature drop across the film under investigation is necessary. The bulk temperature of the liquid inside the evaporator is measured by a travelling thermocouple and the temperature of the tube wall, very near the film, at various levels are measured by fixed thermocouples. The installation of these thermocouples in the tube wall is a skilful and at the same time troublesome job. This laborious method has been avoided by finding out the combined annular film and wall coefficients, the value of which has been used in finding out the individual film coefficient from the general equation of the interrelationship between the overall heat transfer coefficient and the individual coefficients. An almost constant high value of the annular film coefficient has been assured by passing hot water through the outer annulus at a constant temperature and a constant high velocity. Boiling film coefficients of water in natural circulation vertical evaporator, with stainless steel tube and copper tube under low pressure and in Kestner evaporator at atmospheric pressure have been determined. Boiling /-

Boiling film coefficients of organic liquids, ethyl alcohol and toluene, have been determined at low pressure in copper tube natural circulation vertical evaporator. In a forced circulation copper tube vertical evaporator the film coefficients of water, alcohol and toluene have been determined at boiling points below the atmospheric. All the observed film coefficients in different evaporators have been correlated with dimensional equations. In a pyrex glass tube natural circulation vertical evaporator the nature and the mechanism of boiling have been studied with water, alcohol, toluene and a solution of a frothing agent in water when the feed liquids are both saturated with and free from dissolved air. Attempts have been made to determine the latent heat of vapourisation of toluene at different temperatures.

INTRODUCTION.

Practically all the operations that are carried out by the Chemical Engineers involve the production or absorption of energy in the form of heat. The problem of heat transmission is encountered in almost every industry, and because of the diversity in the fields of application there exist countless differences in detail. However, the principles underlying the problem are everywhere the same. The laws governing the transfer of heat, and the types of apparatus that have for their main object the control of heat flow, are therefore of great importance for the efficient application of these principles to the design of heating and cooling equipment.

MECHANISM OF HEAT TRANSFER: It is well understood that the transfer of heat may take place by one or more of the three basic mechanisms:

1. CONDUCTION - When heat flows through a body by the transference of the momentum of individual molecules without appreciable displacement of the particles of the body.
2. CONVECTION - is the transfer of heat from one point to another within a fluid, gas or liquid, by the mixing of one portion of the fluid with another. The motion of the fluid may be entirely the result of differences in density resulting/

resulting from the temperature differences, as in natural convection; or the motion may be produced by mechanical means, as in forced convection.

3. RADIATION - A hot body gives off heat in the form of radiant energy which is emitted in all directions. If matter appears in its path, the radiation will be transmitted, reflected or absorbed. It is only the absorbed energy that appears as heat, and this transformation is quantitative.

However, so far the present work is concerned, conduction and convection play very important roles. Radiation seems to play little part.

CONDUCTION:-

In the majority of cases arising in engineering practice, heat flows from some medium into and through a solid retaining wall and out into another medium. The flow through each is, therefore, but one step in a more complicated process, and the resistance offered by the retaining wall is only one of a series of resistances. The necessity for a clear insight into the mechanism of this process is obvious, and the ability to apply constructively a knowledge of it to many problems cannot be attained until the character and significance of each of the individual steps involved are studied and appreciated.

The basic law of heat transfer by conduction can be written in the form

$$\text{Rate} = \frac{\text{driving force}}{\text{resistance}}$$

The driving force is the temperature drop across the solid, since it is apparent that heat can flow only when there is an inequality of temperature.

Consider a wall of thickness L and an area A. Let the temperature drop across the wall be Δt. At steady state if Q is the heat flow in the time θ, then Q/θ is the rate of heat flow and according to Fourier's law

$$\frac{Q}{\theta} = \frac{k A \Delta t}{L}$$

where k is a proportionality constant and is called the thermal conductivity of the substance. Remembering that Δt is the driving force, it is seen that the resistance is L/kA.

Conduction through fluids rarely occurs in practice except when heat flows through thin films. In these cases, however, the thickness of the film is not known, and therefore the equation cannot be applied. This difficulty is circumvented by the use of film coefficients which will be discussed later. Any body of fluid of appreciable size

through/

through which heat is flowing will develop convection currents of such a magnitude that heat is carried both by convection and by conduction.

CONVECTION:-

The interchange of heat between a fluid and a solid surface with which it is in immediate contact is an engineering problem of major importance. It can be mastered only in the light of clear concepts of the mechanisms involved, but even so the extraordinary complexity of the inter-relationships is at first discouraging. However, there is no other major field in which methods of engineering calculations and design have made greater progress during recent years, owing largely to extensive use of the methods of dimensional analysis.

Consider AB, (Fig. 1) the fixed interfacial surface of contact between a solid G and a fluid EF and the fluid flowing parallel to the surface without change in state. At steady conditions let the temperature of the wall at C be equal to t_w, hotter than the fluid. Heat is flowing from the wall into the fluid and there is a temperature rise of the fluid in the direction EF, resulting in the change of the fluid properties and hence probably in modification of heat exchange between wall and fluid. The heat exchange between surface/

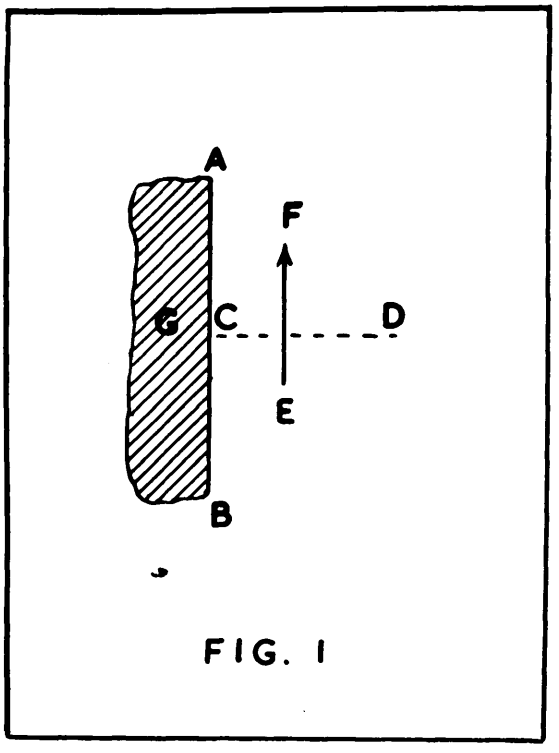


FIG. 1

surface and fluid at this point will be determined, other things being equal, by the surface temperature t_w, irrespective of the mechanism by which this temperature is maintained by energy supply from the left of surface AB. The rate of flow of heat between the solid surface and fluid can be expressed as

$$dq = h\ (t_w - t) = h\ \Delta t\ dA$$

or $dq/dA = h\ \Delta t$, considering dq/dA as a single quantity. (i.e. rate of heat flow per unit contact area). The numerical value of the coefficient h, film coefficient of heat transfer, must ultimately be determined experimentally. The equation is frequently written

$$q = \frac{Q}{\theta} = h\ A\ \Delta t$$

which is independent of point conditions. The value of h is generally expressed as B.Th.U./(hr.)(Sq.ft.)(^{o}F).

It is clear that the numerical value of "t" and hence also of Δt is uncertain because of the necessary existence, along the normal CD, of some sort of temperature gradient in the fluid, which must be hot next the wall and progressively colder farther from it. Generally the temperature t is taken at the bulk temperature of the flowing fluid.

FILM/

FILM CONCEPT:-

When a liquid or a gas is in contact with a solid, there is strong evidence to show the presence of a relatively slowly moving film of fluid on the surface of the solid, a film which becomes thinner as the velocity of the fluid parallel to the surface increases. Through such a film, heat can be transmitted by conduction only, although once the heat has penetrated the film, the hot molecules are picked up and carried away mechanically by the swirling motion of the main body of the fluid, i.e. the transfer is then mainly a matter of convection. Since most liquids and gases are exceedingly poor conductors of heat, one finds a large resistance to heat flow at the boundary of a fluid and a solid, and also at the boundary of a liquid and a gas.

From the mechanism of the transmission, it is not surprising that the coefficient h is by no means constant, even for a given fluid, but is a complex function of a number of variables such as the physical properties of the fluid, the nature and shape of the solid surface, and the velocity of the fluid past the solid boundary. In cases of engineering importance, h varies for different fluids over 20,000-fold. In general, the capacity of heat-transfer apparatus/

apparatus is limited by the thermal resistances of the fluid films rather than by that of the retaining wall, and hence the magnitudes of the individual film resistances become controlling factors in the size of apparatus needed for a given heat-transmission capacity. In all attempts to reduce the boundary resistance between fluid and solid, experience has proved that the most effective means are those tending to reduce the thickness of the surface film. Thus a rapid movement of the body of the liquid or gas past the surface can greatly reduce the resistance by reducing the thickness of the insulating film of fluid.

OVERALL COEFFICIENT OF HEAT TRANSFER:

In the majority of heat-transfer cases met in industrial practice, heat is being transferred from one fluid through a solid wall to another fluid. Let such a case be considered.

C represents the cross section of a solid wall separating the warmer fluid A from the colder fluid E (Fig. 2). B and D are the two fluid films at the two surfaces of the metal wall. The bulk temperature of the warmer fluid is t_1 and that of the colder fluid is t_8. The temperature gradient normal to the surface at the point in question is indicated/

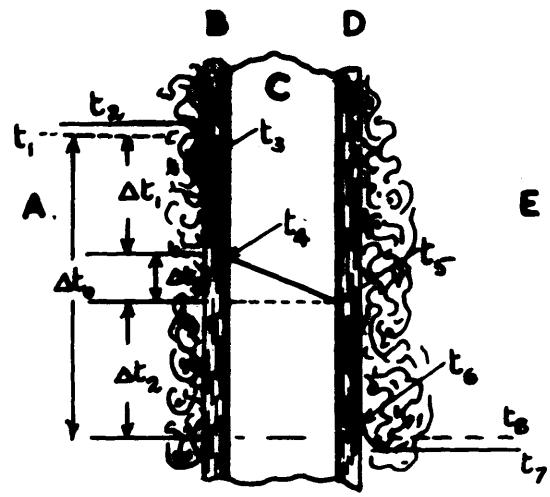


FIG. 2

indicated. Under such conditions Newton found that the rate dq of heat transfer was directly proportional to the overall difference between the temperatures of the warmer and colder fluids $t_1 - t_8$ or Δt_o and to the heat transfer surface dA:

$$dq = U. dA. \Delta t_o \quad ---- \quad (1)$$

and the proportionality factor U is called the overall coefficient of heat transfer and is expressed in B.Th.U./(hr.)(Sq.ft.)(^oF.).

INDIVIDUAL COEFFICIENT OF HEAT TRANSFER:

As a result of the difference in temperature between the hot fluid and wall, $t_1 - t_4$ or Δt_1, the heat flow rate dq through the fluid film B is proportional to Δt_1 and to the heat transfer surface dA_1.

$$dq = h_1 dA_1 \Delta t_1 \quad ---- \quad (2)$$

The proportionality factor h_1 is called the local individual coefficient of heat transfer.

Similarly the rate of heat flow dq through the film D will be

$$dq = h_2 dA_2 \Delta t_2 \quad ---- \quad (3)$$

The/

The rate of heat flow through the wall C will be

$$dq = \frac{k\ dA_3\ \Delta t_3}{L} \qquad \text{---- (4)}$$

where k is the thermal conductivity of the material and L is the thickness of the wall.

If there is any scale deposit on any side of the wall, that should also be taken into consideration.

RELATION BETWEEN OVERALL AND INDIVIDUAL COEFFICIENTS:

Since the rate of heat flow dq is constant and $\Delta t_1 + \Delta t_2 + \Delta t_3 = \Delta t_o$, adding the Eqs. 2,3 and 4:

$$\Delta t_o = dq \left(\frac{1}{h_1\ dA_1} + \frac{L}{k\ dA_3} + \frac{1}{h_2\ dA_2} \right) \text{---- (5)}$$

Comparison of Eq. (5) with Eq. (1) gives:

$$\frac{\Delta t_o}{dq} = \frac{1}{U\ dA} = \frac{1}{h_1\ dA_1} + \frac{L}{k\ dA_3} + \frac{1}{h_2\ dA_2}$$

Where the thickness of the wall is small compared with the diameter of the tube, the relative differences in the various areas involved may be neglected and one may use the equation for series flow through a plane wall

$$\frac{1}{U} = \frac{1}{h_1} + \frac{L}{k} + \frac{1}{h_2}$$

From/

From the discussion of the individual film coefficients, it is clear that the overall coefficient U must be a complex function of operating conditions. Wherever possible it should be evaluated by determination of the individual film coefficients for the case in question, together with the resistances of the intervening solids as determined by their physical properties. While the use of such overall coefficients is allowable in some cases due to extreme convenience, for more rigorous and conservative design of heating and cooling apparatuses more stress should be given to the individual coefficients as they are the controlling factors in the rate of heat transfer.

BOILING AND HEAT TRANSFER TO BOILING LIQUIDS:

To boil a liquid is to generate within it, through the action of heat, bubbles of vapour, which rise and agitate the mass. Here is a process in which heat is being transferred to a substance which is changing phase continually due to the absorbed energy. The process has been known from time immemorial since the first pot was set to boil over a fire but the scientific knowledge of the mechanism of bubble formation and transfer of heat etc. were not known until recently. There are at least two modes of boiling, nuclear and/

and film boiling (22).

When a liquid is being boiled bubbles rise from the heating surface, principally in chains originating at definite points, or nuclei. With the increase in the difference in temperature between the heating surface and the liquid, bubbles begin to rise from numerous scattered points and the rate of heat transfer go on increasing until such a time comes when the number of nuclei is so great that the whole of the surface is covered with a thin film of vapour, and instead of the bubbles rising in regular and neat chains the film periodically bulges and disgorges a large flabby bubble. The rate of heat transfer no longer increases, and if the difference in temperature between the heating surface and the liquid is further increased, there is a sharp decrease. This decrease in the rate of heat transfer was observed by Lang as early as 1888 (39) working with an evaporator for salt water. The temperature difference between the heating surface and the liquid at which the heat transfer rate suddenly falls is called the "critical temperature difference". Below this it is nuclear boiling, which gives place to film boiling at the critical temperature difference. For water this value is about 45°F. and the maximum flux is about 400,000 Btu/hr. sq.ft. (42).

Photographic/

Photographic illustrations of nuclear and film boilings of carbon tetrachloride and ethyl acetate are given by Drew and Mueller (22).

MECHANISM OF NUCLEAR BOILING:

Heat transfer to a boiling liquid is largely determined by the ease of bubble formation. The most comprehensive published study of the mechanism of bubble formation is that due to Max Jakob and his co-workers.(32).

If a liquid and a vapour exist together with no exchange of heat between them, then it is a condition of equilibrium that as many molecules pass from the liquid to the vapour phase as return from the vapour to the liquid. In the process of evaporation true thermal equilibrium does not exist, since heat is being transferred from one phase to the other, and the outward migration of molecules from the liquid is not balanced by the number returning.

The problem arises of the factors which cause bubbles to form in a liquid at its boiling point. Not much is known about the phenomenon. The formation of a liquid-vapour interface requires energy, and energy is also required to extend such a surface once it has formed. In evaporation from a free pre-existing surface no further energy is needed by the surface, because continual evaporation does not increase the/

the surface area. But with a vapour bubble, evaporation into the interior increases the volume of the bubble and hence the surface area. This necessitates the absorption of energy by the interface. For a given increase in volume a greater percentage increase in surface area occurs with a small, newly formed bubble than with a larger bubble. Thus there is a high natural resistance to the formation of bubbles. This surface energy requirement accounts for the fact that the vapour pressure inside the bubble is lower than would normally occur for the same temperature and pressure conditions of the environment.

The liquid surrounding a bubble is therefore slightly superheated, and, as was pointed out by Bosnjakowic (7), it is this superheat which provides the motive force essential to heat transfer. This superheat causes the liquid to vapourise readily into the bubble, not only whilst it is attached to the heating surface but also whilst it is ascending through the liquid.

The effect of dissolved air in the operation was studied by De Luc (49) and in 1772 he propounded a theory which states in precise terms that boiling is initiated and sustained by the bubbles of air which become disengaged from the liquid when heated. The part played by the air is to form a centre of evaporation and give a start to the formation of/

of a vapour bubble. In confirmation of this De Luc found that water from which the air has been carefully expelled by boiling could be heated to a temperature of 234.5°F. without boiling.

The theory of De Luc seems to have the most general acceptance up to the present time. Tomlinson, (49), however, found that a wire gauge cage containing air might be lowered into the liquid without exciting ebullition, provided the cage and air be what he terms chemically clean. It appears then that the speck of dust that the air usually contains are the active agents of bubble formations.

As a bubble leaves the heating surface and rises through the pool, the superheat is consumed in vapourising additional liquid, thus increasing the volume of the bubble. By means of stroboscopic photography Jakob and Fritz (33) made a study of the production, rate of rise, and the rate of growth of bubbles. Their experiments indicate that more heat is transferred to the bubble after it leaves the surface than before, and hence the superheating of the liquid plays an important role.

SPHEROIDAL STATE AND FILM BOILING:

The literature on the spheroidal state is very considerable/

considerable, quite old, obscurely located and somewhat confused but the phenomenon is technically important.

"Briefly, the spheroidal state is a non-equilibrium condition assumed by two or more bodies of solid or liquid when an attempt is made to bring them together while their temperatures differ by more than a definite amount; it is characterized by the subsistence of a layer of vapour between the bodies, which resists and prevents their being brought into contact" (22). When a drop of water is let fall on a sufficiently hot metal plate, the drop will be seen rolling about and wildly dancing on the surface like a globule of mercury. Adequately recorded experimental work dated from a Latin tract by Leidenfrost, published in 1756, but in some of its forms it must have been known from very early times. The laundress's mode of testing the temperature of a smoothing iron by means of a drop of water is an example.

One noticeable observation is the relative slowness of evaporation of the drops even on very hot surfaces. The result of allowing the surface to cool is startling; suddenly the drop flashes into vapour, sometimes with explosive violence. This happens when the surface temperature has fallen below that for which the spheroidal state is possible, and the drop therefore suddenly loses the protecting film/

film of vapour which has been beneath it. Although with drops there is no bubbling this behaviour is essentially analogous to that in boiling when the critical temperature difference is approached from the high side.

Evidently film boiling is an instance of the spheroidal state. Film boiling is considered to comprise all boiling at temperature differences in excess of the critical.

Immediately beyond the critical temperature difference the blanketting film seems to exist, but the vapour-liquid interface is very rough and pulsates irregularly with considerable violence. Possibly there is a local complete collapse of the film at irregular intervals. Such an explanation is consistent with the erratic behaviour with hot wires which was observed in this region by Nukiyama (47). Heat transfer across the vapour layer must be largely by conduction and convection as radiation even at the highest temperature level can account for only 10% to 12% of the values obtained. Kristensen (38) arrived at the same conclusion when he investigated heat transfer from hot surfaces to droplets in the spheroidal state.

Analysis of the existing data on numerical results for heat transfer in boiling shows that there are a number of factors of importance for the rate of heat transfer.

A/

A. NATURE OF SURFACE.

The nature of the heating surface has a profound effect upon the rate of heat transfer. Jakob and Fritz (33) showed that there is a sharp increase in the degree of superheat close to the heating surface, which is greater when the surface is rough; but the degree of superheat in the main body of the liquid is about half that for a smooth surface. Thus a rough surface permits boiling with a lower degree of superheat than a smooth surface. They found that a rough copper surface absorbed air and gave initially much higher coefficients in the region of moderate flux, although with continued boiling the coefficients decreased and approached more closely those for the smooth chromium-plated surface. However, Sauer and co-workers (55) found that the coefficient based on the projected area was increased by cutting grooves in the surface while that based on the actual area was reduced. This result was confirmed by Deutsch & Rhodes with similar experiments (21).

There is disagreement concerning the relative order of merit of the surfaces of different metals from the point of view of heat transfer. Whereas the experiments of Sauer (55) and Cooper (17) showed that iron surfaces are better than copper, Pridgeon and Badger (50) found the reverse to be true. Sauer, however, admitted that he and his co-workers had/

had been unable to repeat their results, owing to uncontrollable variations, which included variations in surface conditions. Bonilla and Perry (11) state that chromium surfaces tend to encourage film boiling, and they attribute this to the very thin film of chromium oxide which forms on all chromium surfaces, and which is not readily wetted. They found that with a temperature difference of 22°C. ethanol gave a heat-transfer coefficient (B.Th.U./hr.) of 4,800 for polished copper, 2,200 for fresh gold plate, 1,300 for fresh chromium plate, and 580 for aged chromium plate.

B. EFFECT OF ADDITION AGENTS:

The direct effect of addition agents is either to change the condition of the surface or the properties of the liquid which ultimately affect the rate of heat transfer. A saturated solution of benzyl mercaptan in distilled water, boiled at atmospheric pressure raised the value of q/A approximately 30% over that for distilled water in the region below that of the critical temperature difference. An excess of mercaptan on a copper tube caused deposition of a yellow scale, lowering the flux at the various temperature differences.

Rhodes and Bridges (52) show the effect of small concentrations of contaminants upon the heat transfer coefficient at/

at a given difference of temperature to water boiling at atmospheric pressure. With a clean steel tube, addition of a film of mineral oil seriously reduces the heat transfer coefficient, and subsequent addition of sodium carbonate gives intermediate values. Jakob and Linke (34, 35) found that for a given flux, for water boiling at atmospheric pressure, heat transfer coefficient was increased 23% by adding a wetting agent (Nekal BX), which reduced the surface tension 45%. Insinger and Bliss (31) found that at a constant flux of 12,800, a wetting agent (Triton W-30) increased the heat transfer coefficient 20% while reducing surface tension 27 per cent.

C. EFFECT OF DIFFERENCE OF TEMPERATURE.

The effect of the difference of temperature has already been mentioned. Clearly one should operate the heat-transfer surface at difference of temperature not greater than the critical value, beyond which the flux decreases. Since in the desirable range the flux and the heat transfer coefficient increase with increase in difference of temperature, one may select either the difference of temperature or the flux as an important variable affecting the heat transfer coefficient. According to McAdams it is better/

better to use the difference of temperature rather than the flux.

D. EFFECT OF TEMPERATURE OR PRESSURE.

Although many industrial processes employ evaporators operating at high pressure it is only in recent years that experimental investigations of the mechanism of boiling have been carried out at pressures much above atmospheric. Experiments of Cryder & Finalborgo (19), Kaulakis & Sherman (37), Braunlich (12), show that a decrease in temperature accompanied by a corresponding decrease in pressure, gave a decrease in heat transfer coefficient at a given difference of temperature but the critical temperature difference remains sensibly constant. Bogart and Johnson (10) using benzene in a high-pressure shell found that at a given difference of temperature an increase in temperature and pressure gave an increase in the heat transfer coefficient all the way from atmospheric pressure to the critical pressure, in spite of a carbonaceous deposit due to the thermal decomposition at excessive difference of temperatures beyond that of the critical. McAdams et al (44) working with platinum wires found that higher pressures resulted in high heat fluxes.

E/

E. NATURE OF THE LIQUID.

There is a considerable volume of published date on the boiling of different liquids. With liquids which wet the heating surface the heat transfer is better than with those which do not. McAdams has given a detailed data for different liquids studied by different workers. At a difference of temperature of 20°F. the value of the heat transfer coefficient of methanol is about 35% of that of benzene and 15% of that of water, all boiling at atmospheric pressure on horizontal plates. Aqueous solutions of salts or organic substances usually give lower coefficients than water at the same pressure, for a given difference of temperature or flux. The individual coefficient of heat transfer is markedly influenced by the properties of the liquid (31).

F. EFFECT OF AGITATION.

In 1902 Austin (1) showed that agitation may increase the heat-transfer coefficients for boiling water in the range of moderate flux. It is curious that these results have not been checked or expanded over the last forty years. In the case of water boiling at 212°F., with no agitation other than that caused by the evolution of bubbles it is known that/

that the heat transfer coefficient increases with increase in temperature difference. This increase may be considered to be due to an increase in agitation caused by more vigorous evolution of bubbles. The effect of velocity increase of liquid in vertical tube evaporators is also to increase the heat transfer coefficient.

G. TYPE OF HEATING SURFACE.

Substantially the same coefficients have been obtained with horizontal and vertical heat-transfer surfaces and the data available indicate little effect of substantial variation in the spacing of horizontal tubes in a bundle (42). Surprisingly good agreement is found between data for small evaporators containing 60 tubes and 1 tube. Some of Jakob & Linke's (34, 35) results on a horizontal plate were almost exactly duplicated by Kaiser (36) in a brew kettle 2,000 times as large.

Available data show no significant change in heat transfer coefficient due to a one hundredfold reduction in diameter of a submerged horizontal cylinder. The data of Nukiyama (47) for water boiled at atmospheric pressure by an electrically heated platinum wire having a diameter of 0.0055 in. gave substantially the same coefficients as those of other observers who used tubes having diameters of 0.5 in. or more.

H/

H. SCALE DEPOSIT.

Scale deposit essentially changes the nature of the heating surface, hence even a thin layer of scale may have a large effect on the overall coefficient. A scale of thickness of 0.0079 in. on copper reduces the overall heat transfer coefficient to 580 Btu. from 820 Btu. in the case of polished copper at a difference of temperature of 18°F. when heat transfer takes place from condensing steam to boiling water (50). A peculiar reverse phenomenon was observed by McAdams (43). When boiling distilled water with a freshly cleaned copper tube, a rate of heat flow of 100,000 Btu./hr. ft^2 was obtained, and after four hours, a value of 400,000 Btu. was reached.

DESIGN PROBLEM.

The design of evaporators is accomplished by the use of overall coefficients which are mainly a matter of experience. It has been long realized that the proper approach to this problem is through the use of film coefficients, but there are practical difficulties.

The simplest equation for the design of an evaporator is:

$$\frac{Q}{\theta} = U.A. \Delta T_T$$

where/

where U is the overall coefficient of heat transfer and ΔT_T is the true mean temperature drop based on the true mean liquid temperature and the steam temperature.

The equation cannot readily be used on account of the difficulty of knowing the true temperature difference. Most designs are based on the overall apparent temperature drop, the temperature difference between the steam in the steam jacket and vapour in the evaporator body, because it can be readily determined. Coefficients based on this definition, the overall apparent coefficients of heat transfer, are largely fictitious because the effect of ignoring boiling point elevation, circulatory system heat losses, temperature changes of the liquid in the tubes, etc. is to show a lower coefficient than the true overall coefficient of heat transfer. Design based on data in this system has been a matter of empirical practice rather than logical development. Even with the determination of the true mean temperature drop, ΔT_T, it will be seen that the true representative value of U cannot be obtained by this method as the value of U depends on the individual coefficients which in turn depend on so many other variables. It is desirable, therefore, to reduce this operation to a logical basis by the use of film coefficients. From the film coefficients/

coefficients can be calculated the true overall coefficient of heat transfer and this overall coefficient can be used for the rigorous and conservative design of evaporators. It is, therefore, desirable to determine the individual film coefficients accurately.

The available data on heat transfer to boiling liquids do not lend themselves readily to theoretical treatment or even to the formulation of empirical generalisations. In recent years many investigators have determined film heat transfer coefficients to boiling liquids but it has been so far impossible to evaluate all of the factors affecting this coefficient in such a way as to obtain any systematic results and as yet no satisfactory general equation has been developed for predicting these coefficients. There are wide variations in the numerical values found and the results are usually applicable only to the type of apparatus used.

From the theoretical standpoint a main difficulty arises in defining accurately the condition of heater surface. Even if such a surface could be accurately defined, it is unlikely that we should be very better off from the design standpoint, since in practice the metal could not be maintained in this condition. The primary needs is perhaps for a standard easily reproducible surface against which other heating surfaces could be compared (7).

Different/

Different aspects of the problem have been tackled by various investigators who designed their own apparatus to suit their own special purposes and studied the mechanism of heat transfer to boiling liquids using both horizontal and vertical surfaces, either flat or cylindrical.

HEAT TRANSFER FROM SUBMERGED SURFACES.

Some of the most thorough investigations on boiling have been done by Jakob and his co-workers, notably Fritz and Linke. Most of their works have been published in German journals but a summary of the work in English has been given by Jakob (32). They have studied the problem from a theoretical as well as practical standpoint, using both horizontal and vertical heating surfaces, many of their results being based on actual photographic measurements. Jakob and Fritz investigated the mechanism of vapourisation of water from a flat, horizontal, electrically heated copper plate, for values of heat flux not exceeding 20,000 B.T.U./Hr./Sq. Ft. and confirmed the speculation of Claasen made in 1902. It was found that rougher surfaces give higher coefficients. Absorbed air on the heating surface was found to increase the rate of vapourisation, but this effect gradually disappeared with continued boiling and returned if the surface was allowed to stand in contact with air.

Comparison/

Comparison of the published data for water and carbon tetrachloride boiled with horizontal and vertical surfaces shows that the horizontal plate gives nearly as good results as the vertical surface for both the liquids at atmospheric pressure.

Sauer, Cooper, Akin, McAdams (55), working in a small submerged tube evaporator studied the variation of overall heat transfer coefficient from steam heated aluminium, copper, chromium-plated and iron tubes to boiling water, benzene, ethyl alcohol, ethyl acetate, methyl alcohol, carbon tetrachloride. Heat transfer rates were higher with iron tubes than with copper tubes, indicating that an increase in the number of vapourisation nuclei more than compensated for the decrease in thermal conductivity.

Cryder & Gilliland (20), made direct measurements of boiling liquid film coefficients of heat transmission by means of an experimental evaporator consisting of an electrically heated brass tube suspended in the boiling liquid. By means of suitable thermocouples, temperatures of both the pipe and liquid were measured at widely varying temperature differences for each of eleven different liquids. Dimensionally sound equations developed correlated the heat transfer coefficients so obtained with the physical properties of the boiling liquids.

Cryder/

Cryder & Finalborgo (19) determined film coefficients for eight different liquids at boiling points both above and below atmospheric pressures using an electrically heated "brass tube". The results obtained by them almost tally with the results obtained by other workers - Linden & Montillon (40), Cryder & Gilliland (20), Williams (61), Jakob & Fritz (33) who used different types of evaporators, horizontal and inclined, and of different materials. The data show that a decrease in boiling temperature accompanied by a corresponding decrease in pressure decreased the film heat transfer coefficients at a given difference of temperature.

Rhodes & Bridges (53) worked in a horizontal cylindrical boiling chamber with a horizontal heating tube through which hot mercury was circulated. Experimental evidence shows that wax, mineral oil and oleic acid promote film boiling and addition of sodium carbonate and sodium chloride restore the nuclear boiling. They came to the conclusion that the wetability of the heating surface by the boiling liquid is an important factor in determining the heat transfer coefficients.

Akin & McAdams (2), determined film coefficients with a single horizontal ¾" nickel-plated copper tube to water and three/

three alcohols boiling at atmospheric pressure and under vacuum. A model evaporator containing 60 nearly horizontal chrome-plated ½ inch copper tubes was used to boil distilled water at atmospheric pressure with overall difference of temperatures ranging from 20° to 100° F. The bundle gave approximately the same results as the single tube evaporator.

Cicheli & Bonilla (16) present a study of the effect of pressure on heat transfer to liquids boiling under pressure. As the pressure approaches the critical, the boiling coefficient increases considerably. They show further that, as the pressure changes, the Delta-T at which nuclear boiling changes to film boiling is different, and that the maximum heat flux is dependent on pressure.

Boiling of Freon-12 inside a small horizontal tube heated electrically was studied by Witzig et al (62) who found that after more than 70% of the liquid vapourised, the coefficient fell off, but below this point the usual rapid variation of heat flux with temperature difference was observed. For the range investigated, the liquid feed rate did not affect the heat transfer, and oil contamination was a minor factor (56).

The effect of extreme variation in Delta-T on the boiling of water from fine horizontal wires were the subjects of/

of two investigations. McAdams et al (44) used four platinum wires of various diameters. They observed ~~that~~ the heat fluxes in the film boiling range well above the maximum possible with nuclear boiling. As the wire diameter increased, the maximum heat flux for nuclear boiling increased to a maximum; the corresponding temperature difference remained constant, and the heat flux at the incidence of boiling decreased. Some preliminary tests at high pressures resulted in extremely high heat fluxes. Faber & Scorah (23) boiled water on nickel, tungsten and chromel A & C wires and also varied the boiling pressure. In general the data paralleled the results obtained by McAdams et al (44) with platinum. Up to a temperature difference of 2000°F. a heat flux in the film boiling region in excess of values obtained by nuclear boiling was not found; a continuous curve over the entire temperature difference range was obtained.

Insinger & Bliss (31) determined the individual heat transfer coefficients from a vertical chromium plated tube to boiling water, aqueous solution of a wetting agent (Triton W 30) and of sugar, carbon tetrachloride and isopropanol at atmospheric pressure and for water a few pressures below atmospheric. They concluded that the individual coefficient of heat transfer to boiling liquid is

(a)/

(a) relatively unaffected by time when contamination is absent,

(b) relatively independent of heating surface, shape and size,

(c) markedly affected by the heat flux density,

(d) markedly influenced by the properties of the liquid,

(e) markedly influenced by the pressure.

These facts, combined with the fact, deduced from literature that diameter is not an important variable, have led to a complex formula which fits the atmospheric data of their own to ± 20% and those of Akin & McAdams, Jakob & Linke, Linden & Montillon, Dunn and Vincent very well (31).

HEAT TRANSFER INSIDE TUBES.

A notable trend of recent years has been the increased use of a long-tube vertical evaporator as a reboiler. Such a unit has a considerable advantage over the old tubular still because of the higher heat transfer coefficients possible and the greater ease of cleaning. In one case the use of such a reboiler decreased the still cost by 50%, a substantial saving particularly because a serious corrosion problem made expensive materials necessary (4).

The/

The type of evaporator under discussion is one that has tubes with a ratio of length-to-diameter of 150 to 200; generally 200 to 1 ratio is accepted as the dividing line for long-tube vertical evaporators (3). There has been no uniformity in the actual sizes used, but, possibly a tube 1.25 in. I.D. and 15 to 20 ft. long is typical. The long tube vertical evaporators are divided into two classes:

(1) Natural circulation

(2) Forced circulation.

(1) LONG TUBE, NATURAL CIRCULATION EVAPORATOR.

The long-tube, natural circulation evaporator has received a good deal of study. In the last few years, there have been many publications that have discussed the various phases of the theory of this evaporator.

The long-tube, natural circulation, evaporator falls into the classification of low level, climbing film machines (13). The most common type is the so-called "Kestner" evaporator. This particular type was developed in Europe in 1899 (Paul Kestner, British Patent 24024, 1899; U.S. Patent 882,322, 1908). For some reason or other, very little interest was shown until about 1930. Due to the re-development of this type of machine in America, entirely independent/

independent of the old Kestner Evaporator, there is a growing tendency to speak of it as the "LTV" (long tube vertical) evaporator. Though generally it is stated that the evaporators are operated with a low liquor level, actually the conception of a static liquor level has no significance in such evaporators.

The characteristic feature of this type of evaporator is the length of the vertical tubes, which are sometimes as long as 22 ft. In operation, the liquid is fed to the bottom of the tubes so slowly that it does not issue at the top as a solid stream, but boils in the tube and is propelled upwards at a rather high velocity by the vapour evolved from the boiling liquid as a spray or a climbing film. In this way the advantage of high velocities, found in forced-circulation evaporators make this type economical and practicable for many kinds of liquids.

A considerable number of papers in foreign literature survey the virtues of the long tube vertical evaporator of the natural circulation type. Among the advantages may be mentioned:

1. Ability to handle thermally sensitive liquids without damage,
2. Relative freedom from scaling trouble,
3. Freedom from foaming.

Within/

Within the last few years, in America, the long-tube vertical evaporator has come into some prominence for handling of materials like glue, sugar syrup, sulphate waste liquor etc. of relatively small unit value but with high viscosities. The latter property makes them difficult to handle in standard types of natural circulation evaporators, while the pumping cost is a disadvantage of the forced circulation evaporator. It has been found that all of these materials, in addition to dilute salt brine, may be handled economically in this type of machine (Swenson Evaporator Co.).

2. LONG TUBE, FORCED CIRCULATION, EVAPORATOR.

It has been recognised from a long time that high liquor velocity in the evaporator is desirable. This is obvious from the facts that the principal thermal resistance in the transfer of heat from a condensing vapour through a metal wall to a boiling liquid is in the liquid film and this film is reduced in thickness by higher velocities.

In this type of evaporator the unevaporated liquid from the vapour-liquid separator is caused to flow rapidly past the heating surface by means of a pump, thus reducing the film thickness and increasing the coefficient of heat transfer. With upward flow in vertical tubes, the vapour generation may be/

be unimportant until near the outlet; in such case most of the surface is used in heating liquid and hence the coefficients approach those for heating liquids in tubes without vapourization. This was found to be the case for data on sugar solutions (41) and a number of organic liquids, including aqueous solutions of methanol (63). With a given solution in a forced-circulation evaporator, the mass velocity of fluid past the surface is the most important variable, but viscosity, thermal conductivity and specific heat are also involved. This evaporator is especially suited for foamy liquids, for viscous liquids, and for those which tend to deposit scale or crystals on the heating surfaces.

NATURAL CIRCULATION.

A very extensive investigation on an evaporator with a single vertical tube is reported by Kirschbaum, Kranz and Stark (38a). In this study, the rate of circulation, the liquid temperature gradient along and across the tube and the tube wall temperature gradient were determined. Curves of the distribution of temperature across the tube show no difference in temperature anywhere across the tube axis until one came so close to the wall that one got into stagnant film. The correlation of their data resulted in an entirely empirical formula which is in no way general.

On/

On the long tube vertical type evaporator, almost without exception, the heat transfer coefficients, either film or overall, are based on the entire length of the evaporator tube. They are usually spoken of as "boiling coefficients" but include the effect of a certain length of the tube in which the liquid is not boiling.

Kirschbaum and his co-workers, however, did realise that: "The way to the development of the final equation probably leads to a division of the heating surface into two parts, one for heating the liquid and the other for evaporation" (13).

A series of important papers dealing with the different aspects of heat transfer in long tube vertical evaporator have been published.

The first of the series is a paper by Brooks and Badger (13). In a single vertical tube of copper 1.76 inch I.D. by 20 ft., jacketted by condensing steam, they measured the distribution of temperature of the liquid along the tube by a travelling thermocouple and determined the boiling and non-boiling sections by the method suggested by Boarts. Studying each section separately they presented a correlation of the overall heat transfer coefficient in the boiling section of the tube. Previous investigators on boiling in vertical evaporator tube had/

had treated the tube as a whole, and coefficients presented were not true boiling coefficients but included the effect of a portion of the tube in which the liquid was being only heated to the boiling point. True overall coefficients for the boiling section were determined for distilled water boiling at temperatures ranging from 150° to 200°F. The overall heat transfer coefficient in the boiling section increased with increase in overall difference in temperature while the overall heat transfer coefficient in the boiling section decreases with increase in fraction of feed evaporated. These workers also erected a glass tube evaporator, approximately $\frac{3}{8}$" O.D., in a glass steam jacket in order to verify the mechanism of boiling stated by Barbet. Regarding the mechanism of boiling three types of action, froth, slug and film had been observed.

The second paper was by Stroebe, Baker & Badger (59) who published work upon the same machine showing the liquid film coefficients obtained when the machine was operated in such a fashion that the non-boiling section was eliminated and the boiling length was constant. Tube wall temperatures were measured by thermocouples, imbedded in the tube wall by the method of Hebbard and Badger (30) at intervals of one foot and the temperature of the liquid in the tube was measured by means/

means of a travelling thermocouple similar to that described by Brooks and Badger. Film coefficients are given for water, sugar and Duponol solutions. The heat transfer coefficients found across the boiling film were higher than expected, and in all cases were considerably higher than the steam film coefficients. From the data obtained, an empirical correlation of the coefficients was derived, expressing the coefficients in terms of the Prandtl number, surface tension of the liquid, specific volume of the vapour, and the average temperature drop across the liquid film, all of these being based on the average temperature of the boiling liquid. A relationship was found between the average liquid temperature and the vapour head temperature, from which it is possible to predict the average liquid temperature, from the variables that are known to the commercial designer.

A third paper by Badger (3) gave a qualitative discussion of the major phenomena that take place in the evaporator tube and the different types of boiling that may occur.

The fourth paper in the series is by Cessna, Lientz & Badger (15). They studied the effect of additional variables, primarily tube diameter and viscosity, in a vertical copper tube, 1.25" O.D., 16 gage, 18 ft. 6½ in. long and calculated true overall coefficients for the non-boiling and the boiling sections/

sections for water and for solutions containing from 30 to 50% sucrose fed at the rate of 36 U.S. gals./hr. The values of overall coefficient for non-boiling section were correlated by the natural convection type equation. Observations indicate that the relationship between the variables and the coefficients in the boiling section is different than that observed in other natural circulation evaporators by other investigators.

A fifth paper in the series was that by Cessna & Badger (14). They repeated the work of Cessna, Lientz & Badger in the same evaporator using a $\frac{3}{4}$" tube. A correlation of data from investigations in which water, 30%, 40% and 50% sucrose solutions were fed to $\frac{3}{4}$", $1\frac{1}{4}$" and 2" tubes at various rates and evaporated at 150° and 200° F. boiling points resulted in a complex equation for the prediction of maximum liquid temperature which is helpful for the design of evaporators used for concentrating heat-sensitive liquids.

Linden and Montillon (40) worked with an inclined evaporator with one heating tube of 1 inch copper pipe about 4 ft. long. The velocity of liquid in the downtake pipe could be measured from the elongation of a phosphor bronze spring. Pipe and vapour temperatures were measured with suitable thermocouples. With distilled water boiling at 180°/

180°, 195°, and 210°F. the overall coefficients were found to increase with the temperature of evaporation and with the difference of temperature.

Foust, Baker & Badger (24) evaporating distilled water in a semi-commercial size basket-type evaporator presented data on liquid velocity and heat transfer coefficients. The correlation of their data resulted in an empirical formula in which submergence ratio, true overall temperature drop, difference in specific volumes of steam and water and the Prandtl number were involved for determining the true overall heat transfer coefficient. It was found that the true overall coefficient of heat transfer and the liquor velocity increased with increase in overall temperature difference and with increase in boiling temperature.

Badger & Lindsay give a review (5) of a Russian paper by Rachko (51). He presents experimental data on the boiling of water inside a vertical tube, and reports no improvement in the coefficient if the velocity through the tube is increased by the admission of steam into the bottom of the tube. The material of construction is unimportant and the ratio of the length to diameter of the tube has no influence on the coefficient, which in effect says that a standard vertical calandria will give the same coefficient as a long-tube vertical machine. A second article revealed that the boiling coefficient/

coefficient is dependent on the heat flux only and is independent of the Reynolds or the Prandtl number. This work was done with an iron tube, the diameter varying from 1.8 to 25 cms. The results of this Russian experiment are at variance with American experience.

Staub (57) presents data on the heat transfer coefficients to boiling water and salt solutions in a Kestner Evaporator. Overall heat transfer coefficients for both boiling and non-boiling sections of the tubes are reported but no indication is given as to the relative lengths of the sections. The unit was run at constant steam pressure with varying inlet temperatures and feed rates of water and salt solutions of different concentrations boiling at atmospheric pressure. Overall boiling film heat transfer coefficients as high as 1440 B.T.U./hr. and vapour velocities out of the tube as high as 120 ft. per sec. are reported.

A paper by Rumford (54) describes experiments with a stainless steel tube 9.1 ft. long by 0.5 in. I.D. used as a vertical evaporator. A travelling thermocouple was used to measure the liquid temperature as it varied with the height of the tube. Hot water at constant rate was pumped through/

through the outer annulus while the liquids were boiled inside the tube at pressures below the atmospheric. Considerable data are given on operation at approximately constant heat flux with varying feed rates of water, ethyl alcohol, toluene and nitric acid. Overall coefficient in the boiling section was calculated in the case of water assuming that boiling took place from the point of maximum temperature of the liquid onwards and using the integrated temperature difference in the boiling zone. Using the previously determined value of wall and annular film coefficient, values for the boiling film coefficient were calculated. The coefficients thus calculated increased rapidly as the feed rate increased, while the total evaporation remained constant. But if the film coefficients were calculated considering the whole length of the tube and the integrated temperature difference for the whole tube the values were fairly constant. Film coefficients thus calculated were given in the case of all the four liquids. Among other conclusions, it was held that the artificial division of a vertical tube evaporator into boiling and non-boiling zones is not justifiable, and that with the evaporation of more than 10% of the feed liquid, the boiling film coefficients are not affected by mass velocities in the evaporator tube.

Part/

Part of tne present work is a continuation of the same work, using the same apparatus, but on an elaborated system.

FORCED CIRCULATION.

Logan, Fragen & Badger (41) worked in a Swenson semi-commercial forced circulation evaporator with twelve 8-foot copper tubes, ¾" I.D. for the purpose of obtaining a correlation of the liquid film coefficients in evaporators by means of an equation involving the physical properties of the liquid being evaporated. In order to obtain a wide range of values of the physical constants, sucrose solutions were used at various concentrations and temperatures. Though in some cases there was some boiling, Dittus-Boelter equations correlated their data very well.

Boarts, Badger & Meisenburg (9) used a single tube forced circulation vertical evaporator, 12 ft. long, 0.76 inch O.D. All heat transfer data were to distilled water boiling at 140° to 212° F., inlet velocity of liquid 2.5 to 15 ft. per sec. and the apparent overall temperature drop, 18° to 72° F. Temperatures along the tube wall were measured by ten thermocouples installed at 15 inch intervals by the method of Hebbard & Badger. The liquid temperature rise in the tube was followed by a travelling thermocouple. Their/

Their data confirmed Logan's claim that Dittus-Boelter equation can predict the liquid film heat transfer coefficient and demonstrated that the predominant mechanism in the usual forced circulation evaporator to be simple heating without boiling. Based on the temperature distribution curves, a formula was evolved for the prediction of average liquid temperature in the tube with a knowledge of the temperature of the liquid at entrance, steam temperature, the velocity of the liquid at the entrance and the length of the tube. An hypothesis, based on the temperature and heat transfer data, visualizes a three part mechanism of simple heating through viscous films, incipient vapourisation, and stable vapourisation by conversion of sensible heat to latent heat.

In a forced circulation vertical heater equipped with two ten-foot 0.75 in I.D. nickel tubes, Ullock & Badger (60) determined the liquid film heat transfer coefficients for petroleum oil and linseed oil at high temperatures from condensing Dowtherm A. All the results correlated very well with the Dittus-Boelter equation as advocated by McAdams. The linseed oil data were even better expressed when the exponents of the Reynolds and Prandtl groups were increased to 0.87 and 0.5 respectively with a value of the constant 0.00725.

It is worthwhile to mention here a comment by Badger who has done most of the fundamental works in this field:- "To present a complete picture of the performance of the long tube, vertical, natural circulation evaporator is so complicated that it is impossible to hope that this can be accomplished for some time to come " (15). So, though much work has been done, much more remains to be done in order to get a clearer picture of the mechanism of boiling and the mechanism of heat transfer to boiling liquids.

SCOPE OF THE PRESENT WORK.

In most experimental work involving the study of the heat transfer coefficient generally the heating medium used is steam. To determine the boiling film coefficient it is therefore necessary to know the value of the steam film coefficient and the wall coefficient. Assumption of the approximate heat transfer coefficient of steam film, even from accepted data, is not justifiable as one cannot be sure of the nature of condensation of the steam, dropwise or film-condensation. Determination of the temperature of tube wall then becomes an integral part in the experimental study of the heat transfer coefficient to and from these surfaces through the enveloping films. The final degree of accuracy obtained in such work is usually controlled by the accuracy of the determination of surface temperature.

By far the major portion of the work done on this problem has been carried with thermocouples, perhaps because of their inherent economy and flexibility of installations. But the satisfactory installation of thermocouples is a very skilful and troublesome job. So, if some other arrangements can be made, which can do away with the installations of thermocouples for the determination /-

determination of tube wall temperatures without seriously affecting the accuracy of the values of the film heat transfer coefficients, it can be considered an advance in this important branch of science.

As shown by Rumford (54) the apparent value of the boiling film heat transfer coefficient increases rapidly with the increase of feed rate and ultimately reaches unreal value. This cannot be correct and a more complete determination of boiling film heat transfer coefficients for different liquids, in tubes of different materials, was attempted. It was also proposed to make a visual study of the mechanism of boiling in a pyrex glass long tube vertical evaporator and forced circulation evaporation problems were also undertaken. It was hoped to correlate the results with equations involving the physical properties of the various liquids.

The latent heat vapourisation of toluene at different temperatures were not known to have been given in detail by any previous worker. As toluene was one of the test liquids, it was decided to determine the latent heat of vapourisation of toluene at different temperatures.

PART I

NATURAL CIRCULATION VERTICAL EVAPORATOR.

PART I

SECTION A

STAINLESS STEEL TUBE EVAPORATOR.

PART I.

SECTION A.

(a) DETERMINATION OF THE COMBINED ANNULAR FILM AND WALL COEFFICIENTS.

In a long tube vertical evaporator, if a heating fluid is passed through the annular part and a feed through the central tube, it is easy to find out the overall heat transfer coefficient from the hot fluid to the colder one from the general equation

$$U = \frac{Q}{A \cdot \Delta T}$$

where

U - Overall heat transfer coefficient, B.Th.U./hr.

Q - Quantity of heat in B.Th.U./Hr./Sq.ft.

A - Heating area, sq.ft.

ΔT - Difference in temperature between the hot and the cold fluids.

The overall heat transfer coefficient is made up of three other coefficients and the relation between them is given by

$$\frac{1}{U} = \frac{1}{H_A} + \frac{1}{H_W} + \frac{1}{h}$$

where/

where

U - Overall heat transfer coefficient, B.Th.U./Hr.

H_A- Film coefficient of Heating Medium, (Annular film coeff.)

H_W- Wall coefficient,

h - Film coefficient of the feed liquid.

For determining any individual coefficient, a knowledge of the other two coefficients is necessary if it has to be determined from the overall heat transfer coefficient. It is also possible to determine the individual film heat transfer coefficient directly if the difference of temperature across the fluid film in question can be determined. Thus individual film coefficients of heat transfer may be determined either by installing thermocouples to measure the temperature of the pipe wall in contact with the film in question or by arranging conditions so that all thermal resistance except that of the film in question remains constant.

A liquid heating agent flowing at high velocity and constant average temperature through the annulus enables the film coefficient of the heating agent to be maintained at a high constant value. Thus for a definite temperature and velocity of the annular stream, it is a justifiable assumption that/

that the resistances of the outside film and of the wall are essentially constant (26). McMillen & Larson (45) utilised hot water, circulating at a constant average temperature of 150°F. and a constant rate of 20 gal/min. through the inner pipe, as a heating agent for the determination of heat transfer coefficient for turbulent flow.

Using the same principle of McMillen & Larson (45) hot water at a constant temperature and a constant high rate was circulated through the outer annulus and water at different known velocities were circulated through the inner tube in order to find out the combined value of the annular film coefficient and the wall coefficient in the following way.

h can be determined from the Dittus-Boelter equation for liquids flowing inside tubes (42).

$$h = 0.0225 \frac{K}{D} \left(\frac{DVP}{\mu}\right)^{0.8} \left(\frac{\mu C}{K}\right)^{0.4}$$

where

K - Thermal conductivity of the fluid, B.Th.U./(Hr.) (Sq.ft.) (°F./ft.)

V - Velocity of fluid, ft./sec.

D - Diameter of tube, ft.

P - Density of liquid, lbs./cu.ft.

C - Specific heat of fluid, B.Th.U./lb/°F.

μ - Viscosity of fluid, F.P.S. unit.

With/

With the exception of viscosity, the properties of the fluid comprising the film vary only slightly with temperature. Therefore, in the above equation, K, D, ρ and C can be taken as constants and the equation can be written as

$$h = K' \frac{v^{0.8}}{\mu^{0.4}}$$

If 1/U is plotted against $\mu^{0.4}/v^{0.8}$, (58) the points should lie on or about a straight line and if the straight line is drawn back to cut the Y-axis, the value of 1/U when the value of $\mu^{0.4}/v^{0.8} = 0$, i.e. when the velocity of feed liquid is infinitely fast, is obtained. Under this condition the film coefficient h will also be infinite or $1/h = 0$. This means that

$$\frac{1}{U} = \frac{1}{H_A} + \frac{1}{H_W}$$

This intercept gives the sum of the pipe wall and annular fluid film resistances. Once this combined value has been found out, it can be directly used for finding out the value of film coefficient in the boiling section of the evaporator tube in the later experiments.

McMillen & Larson plotted 1/U vs. $\mu^{0.5}/G^{0.8}$ but from the equation advocated by McAdams it can be found that the best method is to plot 1/U vs. $\mu^{0.4}/v^{0.8}$.

DESCRIPTION/

DESCRIPTION OF APPARATUS.

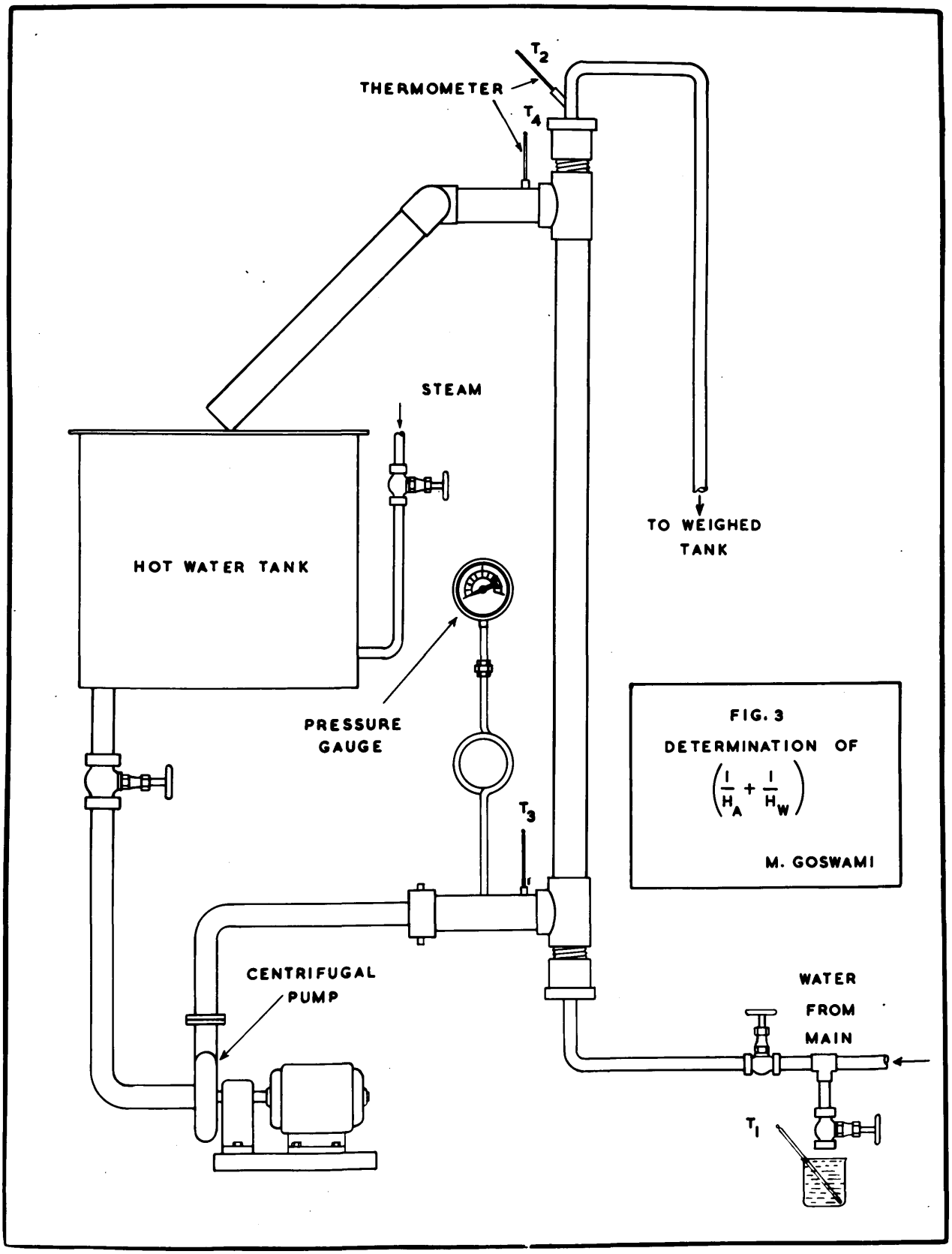
A working diagram of the apparatus is shown in fig. 3.

The heating tube consists of an inner stainless steel tube (Staybrite F.M.B.) 0.7" O.D. and 0.5" I.D., and heated by a hot water jacket for 9' 2" of its length. This gives a ratio of length-to-diameter of 220. The hot water jacket consists of an iron pipe ($1\frac{3}{8}$" O.D. and 1" I.D.) with iron Tee-pieces screwed on at either end. These latter act as inlet and outlet for hot water. Two thermopockets, one in each Tee-piece, permit of the temperature readings.

Two brass stuffing boxes, each with caps and gland nuts, situated at the top and bottom of the hot water jacket, and packed with asbestos cord provide the means of holding the inner tube in position.

Hot water from a tank, heated by open steam, is circulated through the hot water jacket, co-current with the feed water. The hot water from the tank is circulated by a centrifugal pump and enters by 1" I.D. flexible copper hose joined to the bottom Tee-piece and leaves by a long side arm joined to the top tee-piece. The pressure of the water registers on a pressure gauge fitted to the bottom tee-piece.

The/


T_2
THERMOMETER
T_4
STEAM
HOT WATER TANK
TO WEIGHED TANK
PRESSURE GAUGE
T_3
CENTRIFUGAL PUMP
WATER FROM MAIN
T_1
FIG. 3
DETERMINATION OF
$\left(\frac{1}{H_A}+\frac{1}{H_W}\right)$
M. GOSWAMI

The inner stainless-steel tube is fitted on to a ½" I.D. copper pipe by thick rubber tubing. The copper pipe, which has a thermometer pocket, takes the feed away to waste or to a weighed tank as may be desired.

Four thermometers required for measuring the temperatures were standardised against an N.P.L. standard thermometer.

EXPERIMENTAL PROCEDURE.

The heating water was pumped through the annular space as fast as possible. The maximum pressure obtained was about 15 lbs/sq.in. but it fluctuated to a great extent. The pressure was fairly constant at about 13 lbs. per sq.in. and it was tried to keep this pressure throughout the tests.

The feed water was admitted from a tap and to ensure that this rate remained steady, a main water tap was left open during all tests.

The plant was allowed to run for a considerable period of time to ensure steady conditions and the heated feed water was run to waste. At steady conditions when all the thermometers showed steady temperatures, the feed exit was switched to a weighed tank and the following readings were observed at an interval of one minute.

(1) Inlet temperature of the heating water

(2)/

(2) Outlet temperature of the heating water

(3) Inlet temperature of the feed water

(4) Outlet temperature of feed water

(5) Pressure of the heating water.

The rate of flow of the heating water was also measured by collecting the water in a bucket for a known interval of time and weighing it. The velocity of flow through the annular space was about 10 ft./sec.

At the end of 10 minutes the feed was again switched to waste and the amount of water collected in 10 minutes was weighed.

The above procedure was repeated with increasing feed rates.

Two different sets of observations were recorded at two different temperatures of the heating water at 50° C and 80° C.

CALCULATION OF RESULTS.

(1) The heat load was found out as follows:

Suppose the rate of feed is x lbs/hr.

T_1 - Temperature of feed water entering, in °F.

T_2 - Temperature of feed water exit, in °F.

Then heat load (Q) = $x(T_2 - T_1)$ B.Th.U./hr.

(2)/

(2) The logarithmic mean temperature difference was calculated as follows:

T_3 - Temp. of heating water entering jacket, in °F.

T_4 - Temp. of heating water leaving jacket, in °F.

then

$$\text{log mean temp.diff. } \Delta T = \frac{(T_3 - T_1) - (T_4 - T_2)}{2.3 \log_{10} \frac{T_3 - T_1}{T_4 - T_2}}$$

(3) The heating area of the tube was calculated using the arithmetic mean of outer and inner diameters:

I.D. 0.5")
) Mean 0.6"
O.D. 0.7")

Length - 9' 2"

Therefore, heating area is 1.44 sq.ft.
Area of cross section = 0.001364 sq.ft.

(4) The overall heat transfer coefficient in B.T.U/hr./sq.ft./°F. was obtained thus:

$$U = \frac{\text{Heat load } (Q)}{\text{Area } (A) \times \text{log mean temp. diff. } (\Delta T)}$$

where U is the overall heat transfer coefficient.

(5) V was found out by converting the feed rate of lbs/hr. to cu.ft./sec. and dividing this by the cross-sectional area of the tube.

(6)/

(6) ρ was taken to be 62.3 lbs/cu.ft. in all cases.

(7) μ was taken at the arithmetic mean of inlet and outlet temperatures.

(μ in poise x 0.0672 = μ in F.P.S. units)

The value of $\frac{(v)^{0.8}}{(\mu)^{0.4}}$ was calculated for each observation. The results are given in Tables 1 & 2.

For two sets of results at 50° C and 80° C of heating water, two graphs were drawn with 1/U against $\frac{(\mu)^{0.4}}{(v)^{0.8}}$. The two graphs are given (Figs. 4 & 5). The values of 1/U when $\frac{(\mu)^{0.4}}{(v)^{0.8}}$ is zero or in other words when V (velocity of feed) and consequently h are infinite and 1/h is zero are found from the graphs. The values are

At 50° C (Heating Water) - 0.0015

At 80° C (Heating Water) - 0.0013

As the value of the intercept decreases, the value of $1/H_A$ decreases since the value of the wall resistance remains practically constant. In other words, with the decrease of the value of the intercept, the value of the annular film coefficient, H_A, increases. McMillen & Larson got a value of approximately 2500 B.Th.U./Hr. for the heating water at 150°F. when the water was being circulated at a constant rate of 20 gal/min. through the pipe.

TABLE - 1

DETERMINATION OF $\left(\frac{1}{H_A} + \frac{1}{H_W}\right)$

Total heating area - 1.44 sq.ft.

Test No.	Feed water In T_1	Feed water Out T_2	Heating water In T_3	Heating water Out T_4	ΔT ^{o}F	Feed lbs/hr.	Vel. ft/sec. V	$\frac{Q}{A}$ B.Th.U./hr.	$1/U$	$\mu \times 10^4$ F.P.S.	$\frac{DV\rho}{\mu}$	$\frac{V^{0.8}}{\mu^{0.4}}$	$\frac{\mu^{0.4}}{V^{0.8}}$	U B.Th.U./hr.
1.	52.03	83.59	122.07	121.64	52.50	270	0.8824	8521	0.008876	6.770	4061	16.76	0.05966	112.70
2.	53.02	86.99	122.43	121.64	50.08	278	0.9069	9428	0.007653	6.594	4284	17.32	0.05773	130.70
3.	52.52	88.23	123.15	122.09	50.07	308	1.005	10980	0.006566	6.540	4785	19.52	0.05123	152.30
4.	53,11	88 .11	122.00	120.42	48.37	348	1.137	12180	0.005718	6.524	5431	20.84	0.04818	174.90
5.	53.78	89.78	122.59	121.59	46.72	354	1.157	12880	0.005322	6.423	5610	21.26	0.04730	187.90
6.	51.98	87.76	122.09	120.50	49.14	360	1.177	12880	0.005495	6.586	5569	21.34	0.04686	182.00
7.	52,83	87.80	122.95	122.19	50.21	405	1.323	14170	0.005104	6.549	6296	23.50	0.04255	195.90
8.	52.81	86.54	121.96	120.07	49.25	437	1.427	14720	0.004817	6,594	6738	24.87	0.04020	207.60
9.	50.79	84.38	122.00	119.89	51.36	489	1.598	16430	0.004503	6.787	7333	26.93	0.03713	222.10
10.	51.01	83.44	122.11	119.43	51.63	548	1.789	17750	0.004188	6.826	8168	29.41	0.03400	238.80
11.	50,50	81.25	123.55	120.38	54.41	722	2.356	22170	0.003534	6.949	10560	36.39	0.02748	283.00
12.	48.29	80.15	122.72	119.77	55.26	714	2.333	22750	0.003499	7.115	10210	35. 76	0.02794	285.80
13.	48.29	78.29	122.41	119.34	56.04	801	2.618	24030	0.003359	7.211	11310	39.00	0.02564	297.70
14.	48.29	77.86	123.01	119.87	56.86	834	2.726	24670	0.003320	7.231	11750	40.26	0.02484	301.20
15.	48.29	77.00	123.28	119.95	57.55	918	3.000	26360	0.003145	7.278	12840	43.34	0.02307	318.00
16.	50.32	75.52	122.90	119.05	56.93	1026	3.353	25850	0.003171	7.246	14420	47.45	0.02108	315.00
17.	49.75	71.29	122.72	118.36	59.14	1380	4.510	33970	0.002508	7.449	18740	59.33	0.01683	398.80

TABLE - 2

Heating Water at 80° C.

Total Heating Area - 1.44 sq.ft.

Test No.	Feed water In T_1	Feed water Out T_2	Heating water In T_3	Heating water Out T_4	ΔT °F.	Feed lbs/hr.	Vel. Ft/sec. V	Q B.Th.U./hr.	U B.Th.U./hr.	$1/U$ x 10^3	$\mu \times 10^4$ F.P.S.	$\frac{DV\rho}{\mu}$	$\frac{V^{0.8}}{\mu^{0.4}}$	$\frac{\mu^{0.4}}{V^{0.8}}$
1.	56.39	150.49	178.03	178.84	64.14	153	0.500	14440	155.9	6.414	4.434	3514	12.60	0.07937
2.	56.03	147.52	175.32	175.69	61.77	198	0.647	18115	203.7	4.904	4.514	4466	15.38	0.06502
3.	53.42	127.94	176.80	175.60	79.68	351	1.148	26160	228.0	4.386	5.098	7011	23.16	0.04318
4.	55.63	126.32	176.97	175.86	80.27	404	1.319	28530	246.8	4.052	5.081	8087	25.93	0.03856
5.	53.78	121.46	174.88	173.32	81.80	468	1.529	31660	268.8	3.720	5.279	9036	28.75	0.03478
6.	53.06	118.90	175.42	173.83	83.75	513	1.677	33780	280.0	3.571	5.381	10160	31.27	0.03198
7.	49.01	102.74	176.71	172.22	95.85	870	2.843	46740	338.6	2.964	6.091	14500	44.59	0.02252
8.	51.98	100.24	174.20	170.25	93.87	1008	3.294	48570	359.3	2.784	6.075	16890	50.20	0.01992
9.	49.51	93.92	175.73	169.34	98.79	1230	4.020	54630	384.0	2.604	6.437	19450	57.51	0.01739
10.	49.10	92.98	176.72	170.06	100.20	1326	4.330	58190	402.6	2.492	6.485	20820	60.91	0.01642

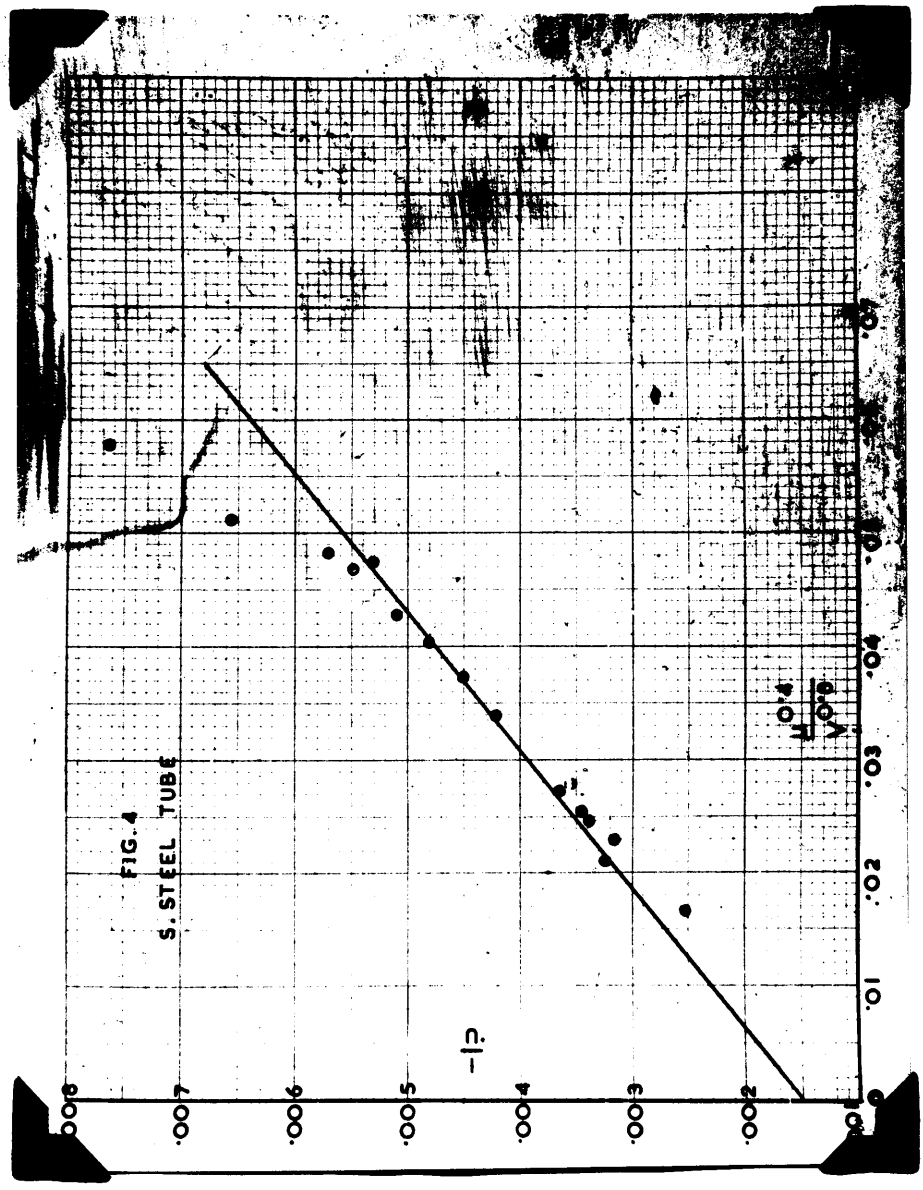


FIG. 4
S. STEEL TUBE
-dp
.007
.006
.005
.004
.003
.002
.01
.02
.03
.04
.05
.06
.07
.08
.09
.1
μ 0.4
V 0.6

FIG. 4

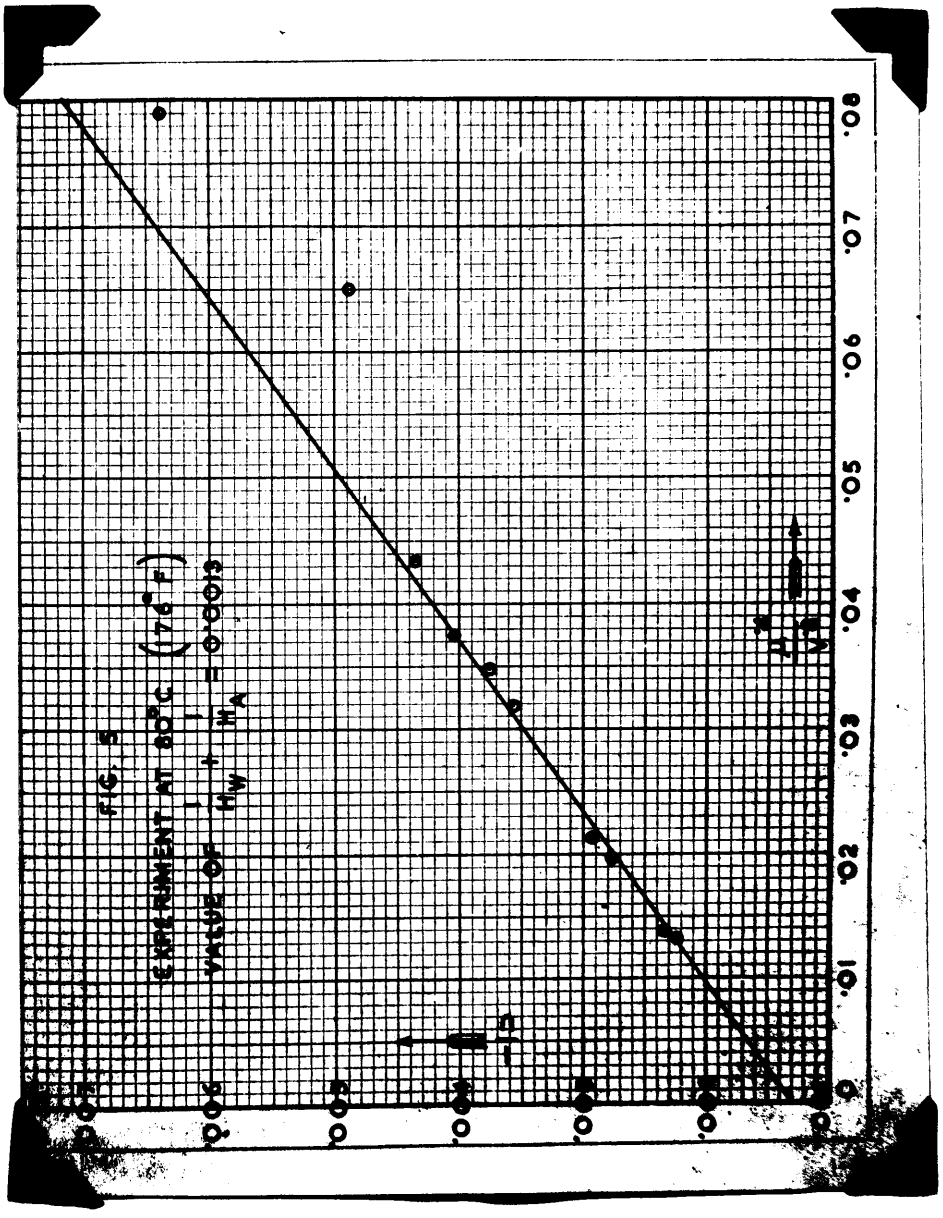


FIG. 5
EXPERIMENT AT 80°C (176°F)
VALUE OF
0.08
0.07
0.06
0.05
0.04
0.03
0.02
0.01
0.0

FIG. 5

PART I.

SECTION A.

(b) DETERMINATION OF THE BOILING FILM COEFFICIENT FOR WATER.

SUMMARY:

The apparatus was fixed up as shown in Fig. 6 and placed under vacuum. Hot water, maintained at a constant inlet temperature of 80° C. (176° F.) was pumped round the annulus as before.

Feed water was sucked in by the vacuum and was partially evaporated during its passage up the inner stainless steel tube. The unevaporated water was collected in a graduated vessel from a liquid-vapour separator while the evaporated water was condensed and also collected in a graduated vessel. During each trial at different feed rate, temperature readings were taken at predetermined heights up and down the inner steel tube by means of a travelling thermocouple.

The evaporator tube was divided into boiling and non-boiling sections. Overall heat transfer coefficients considering the entire length of the tube and the boiling section of the tube were calculated. The boiling film coefficients were also calculated.

DESCRIPTION/

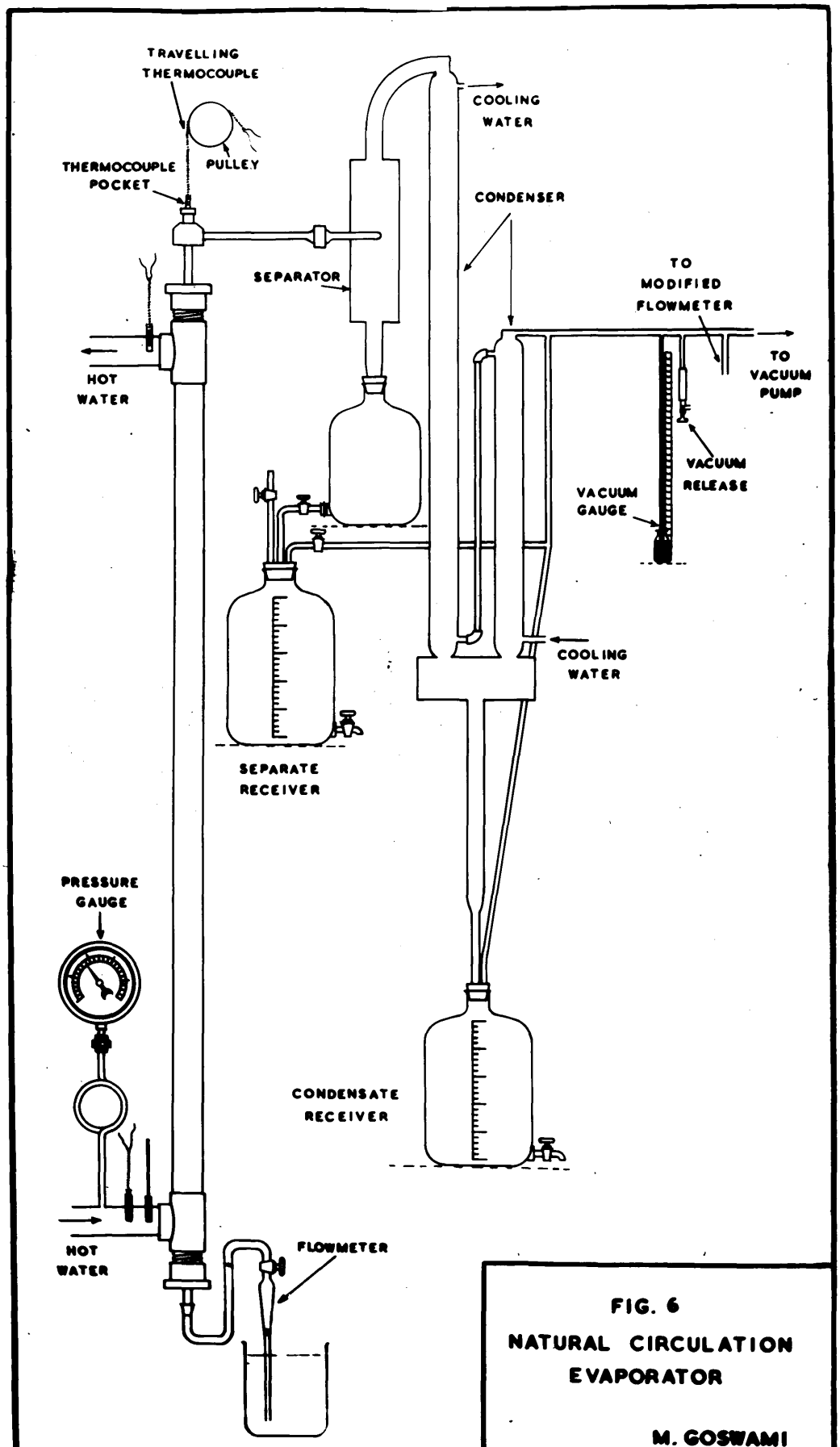


FIG. 6
NATURAL CIRCULATION EVAPORATOR

M. GOSWAMI

DESCRIPTION OF THE APPARATUS.

The heating tube is the same as before but the inner stainless steel tube is screwed into a brass gland piece at the top. This gland piece has a stainless steel side-arm terminating in a flush half-union.

The lead to separator has another flush half-union and the two half-unions are joined by a union nut.

The separator is a stainless steel cylinder (14" long x 3¼" o.d.). The pipe, from the union mentioned above, is welded tangentially to the body of the separator.

A stainless steel pipe (5½" long x 1 1/16" o.d.) is welded at the base of the separator and drains off the "separate".

A curved 1 2/10" pipe, which leads the vapour to the condenser, is welded to the top of the separator and to the top of the condenser making a complete unit.

The condenser consists of:

(a) a stainless steel tube (5' 11" long x 1¼" o.d.) down which vapour from the separator passes. This tube is cooled by a water jacket (5' 7" long x 1⅝" o.d.) welded to the inner tube.

(b) a stainless steel tube (3' 0½" x 1¼" o.d.) up which the vapour from the previous limb passes. This tube is cooled by/

by a water jacket (2' 11½" long x 1⅝" o.d.) welded to the inner tube.

The lower ends of the inner tubes of the two limbs are welded into a cylindrical bottom-piece (7¼" long x 3¼" o.d.). A stainless steel tube (5" long x ¾" i.d.) is welded to the base of the bottom-piece and drains off the "evaporate".

Cooling water enters the water jacket of the smaller limb at the bottom and is conveyed by a ½" o.d. glass tube from this limb to the other limb. The cooling water leaves from the top of the second water jacket. Stainless steel pipes (2½" long x ½" o.d.) provide the means of entrance and exit for the cooling water. A ½" o.d. tube welded to the end of the inner tube on smaller limb provides a vacuum lead.

There are two receivers - the "separate" receiver and the "evaporate" receiver.

The "separate" receiver is made up of two vessels namely, a 2½ litre aspirator vessel and a 5½ litre aspirator vessel, joined by glass tubing and a stop-cock. The 5½ litre vessel is graduated in 50 c.c. divisions and has a lead to vacuum and a lead to the atmosphere. In this way the 5½ litre vessel can be emptied while the smaller vessel continues to receive the "separate".

The/

The "evaporate" receiver is a 15 litre aspirator vessel graduated in 100 c.c. divisions. It is connected to the outlet pipe from the condenser by a glass tube and a rubber connection. A vacuum lead is provided to this receiver also.

The vacuum gauge is a metre of glass tube (3/16" o.d.) connected by a Tee-piece to the main glass vacuum lead. This glass tube is clamped vertically with the lower end dipping into a small bottle of mercury. A metre stick clamped with its zero level on the surface of the mercury, provide a means of reading the vacuum. Glass tubing and rubber tubing provide the connections between the main vacuum lead and the condenser. A vacuum release valve is also provided.

The feed enters the inner stainless steel tube by a glass flow-meter, provided with a stop-cock. A ground glass connection on the flow-meter joins it to the tapered end of the evaporator tube. The flow-meter is provided with a glass bead the position of which varies with the feed rate. A bucket of water, into which the open end of the flow-meter dips, provides the feed reservoir.

The general set of the apparatus can be seen from Fig.6.

CONSTRUCTION/

CONSTRUCTION OF DIFFERENTIAL THERMOCOUPLE.

An arrangement was made for the thermocouple network to measure the difference of temperatures between (1) the inlet heating water and the liquid in the bottom half of the stainless tube, and (2) between the outlet heating water and the liquid in the top half of the stainless tube.

Two 46 ft. lengths of japanned copper and eureka wires (24 S.W.G.) were bared for ½" at one end, twisted together on the bared portion and hard soldered with silver solder on the twisted portion. This soldered end was gripped in a hand drill and the two free ends of the wires held together at wire length. The drill was started and the wires were twisted together. This procedure makes the wires more rigid, prevents "kinking" and makes it much easier to slide the wire in and out of the thermocouple pocket of 3/32" o.d. glass tube which runs throughout the length of the stainless steel tube.

Two other thermocouples were made in the same way with 12 ft. lengths of japanned copper and eureka wires of 24 S.W.G.

The arrangement of the Differential Thermocouple network is shown in diagram 7.

The three free ends of eureka of the three thermocouples were/

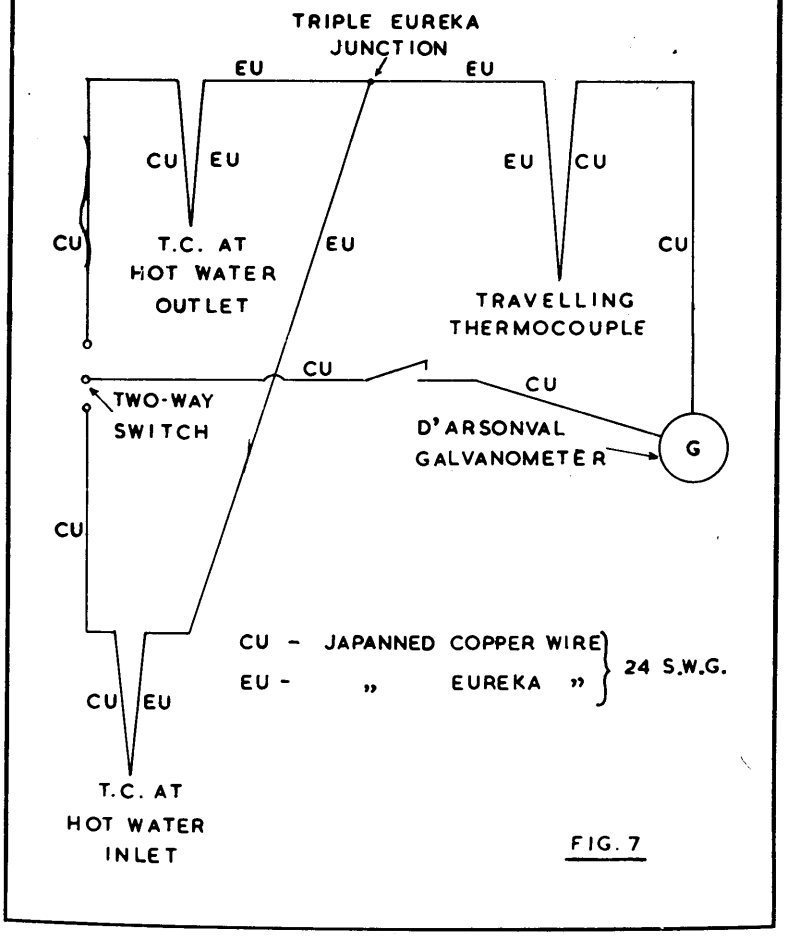

TRIPLE EUREKA
JUNCTION
EU
EU
CU
EU
EU
CU
CU
T.C. AT
HOT WATER
OUTLET
EU
CU
TRAVELLING
THERMOCOUPLE
CU
CU
TWO-WAY
SWITCH
D'ARSONVAL
GALVANOMETER
G
CU
CU - JAPANNED COPPER WIRE
EU - " EUREKA "
24 S.W.G.
CU
EU
T.C. AT
HOT WATER
INLET

FIG. 7

FIG. 7

were hard soldered together to form a triple eureka junction. The copper wire of the travelling thermocouple is directly connected to a D'Arsonval mirror galvanometer. The other two thermocouples, which are to measure the inlet and outlet temperatures of the heating water have their copper wires connected to opposite terminals of a 2-way switch. Another copper wire connected to the middle terminal of this switch goes through another switch to the second terminal of the galvanometer.

In this way the travelling thermocouple can be matched against either the "inlet" or "outlet" thermocouple and the galvanometer can be switched out of the circuit when desired.

On a wooden platform, free from vibrations of any kind, was placed a sheet of felt on which were placed three large rubber bungs. The galvanometer was placed on these rubber bungs and levelled by means of levelling screws and spirit level. A scale was placed at a distance of 125 cms. from the mirror of the galvanometer and at right angles to the beam of light. The lighting arrangement was made from a 6 volt car battery and an electric bulb situated inside a focussing telescope through a switch. The light focussed by the telescope gives a sharp reflection from the galvanometer mirror on the graduated screen just above the telescope.

CALIBRATION/

CALIBRATION OF THE THERMOCOUPLES.

The thermocouples which were to measure the temps. of the "inlet" and "outlet" heating water were made into spiral coil for about 6" from the junction upwards and the coils were placed inside $\frac{1}{4}$" i.d. x $2\frac{1}{2}$" long glass tubes containing oil. The tubes were inserted in a beaker of water kept at 80° C throughout the test. The travelling thermocouple was inserted in a 1 ft. length of 3/32" i.d. glass tube and inserted in a beaker of water kept at temperatures from 55°C to 75°C.

Due to the difference in temperature at the two junctions of the thermocouples, there was a current produced and hence deflection in the galvanometer. The deflections at various temperatures were noted, temp. corrections made and graphs were drawn of difference in temperatures against deflections in centimetres.

The same were repeated, the thermocouples at the "inlet" and "outlet" of the heating water being at 50°C and the travelling thermocouple being at temperatures from 25° C to 47° C. Graphs were drawn in these cases also. The graphs are shown in Figs. 8 and 9.

INSERTION/

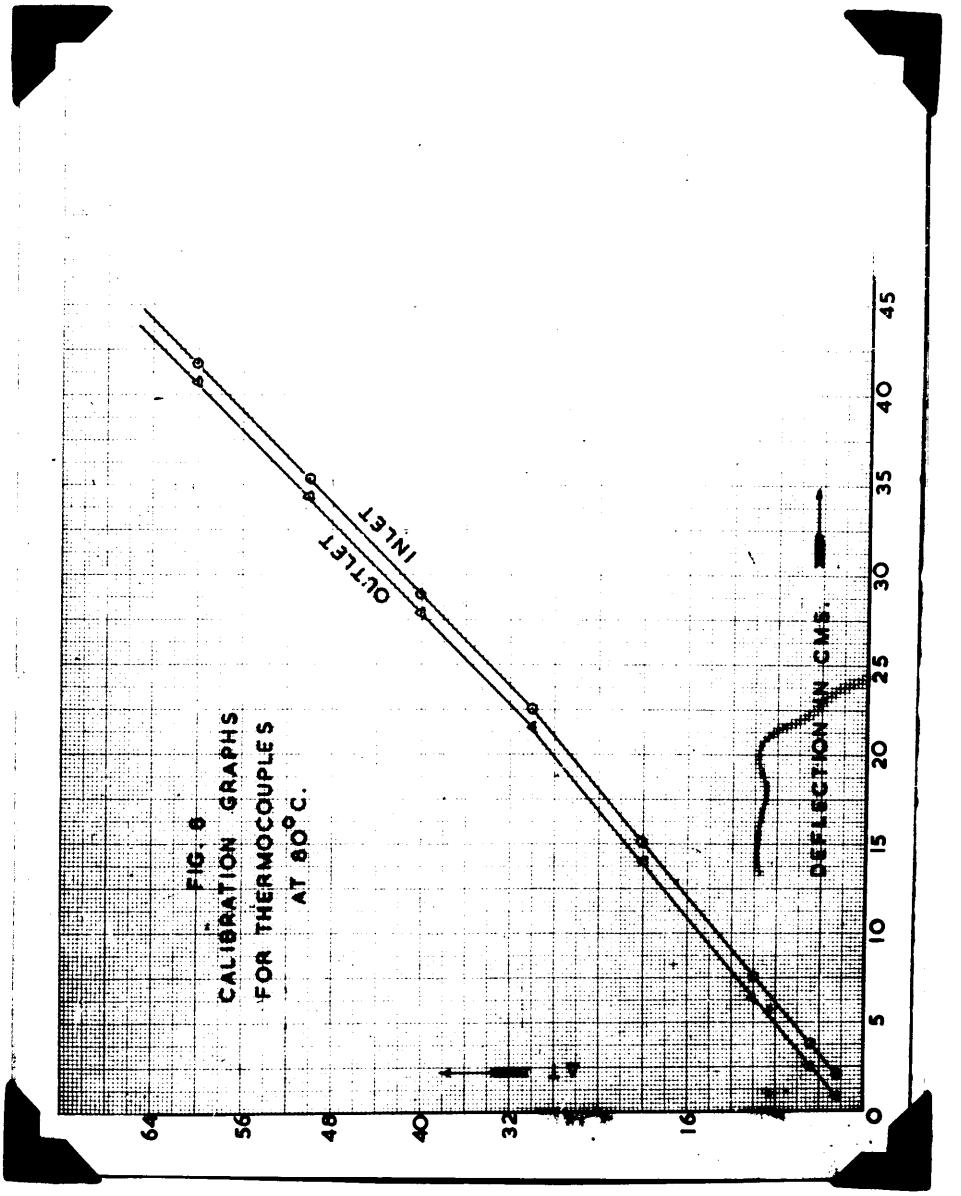


FIG. 8
CALIBRATION GRAPHS
FOR THERMOCOUPLES
AT 80°C.
OUTLET
INLET
DEFLECTION IN CMS.
0
5
10
15
20
25
30
35
40
45
16
32
40
48
56
64

FIG. 8

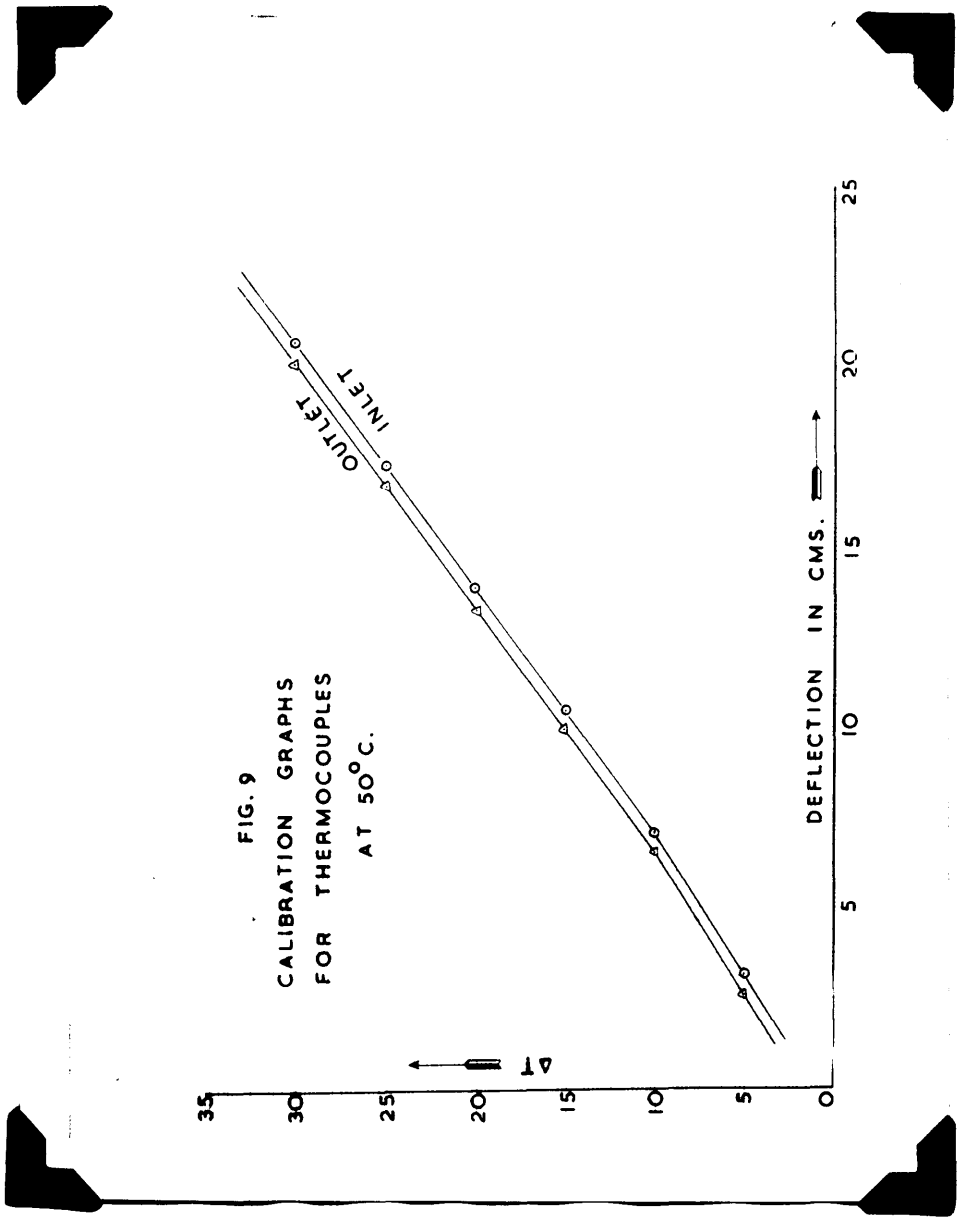


FIG. 9
CALIBRATION GRAPHS
FOR THERMOCOUPLES
AT 50°C.
OUTLET
INLET
ΔT
DEFLECTION IN CMS.
0
5
10
15
20
25
30
35

FIG. 9

INSERTION AND GRADUATION OF THERMOCOUPLE.

The couple was inserted fully down the glass tube and a piece of adhesive tape placed round the wire near the top of the glass tube. A length of string tied round this tape and passing up round a pulley enables the thermocouple to be raised or lowered without fear of "kinking".

The thermocouple was now painted with white paint at the open end of the glass tube. It was then painted at intervals so that the thermocouple end can be placed to a definite position in the tube to read the temp. there.

EXPERIMENTAL PROCEDURE.

The apparatus having been set up as shown in the flow-sheet, hot water at 176°F was circulated through the heating jacket as before. The stop-cock on the flowmeter was closed, the cooling water to the condenser turned on and the vacuum applied (a Genco vacuum pump). A vacuum of 65 cms. of mercury was registered.

The stopcock on the flowmeter was nowopened and feed introduced into the apparatus at constant rate.

The separated liquor or "separate" collected in the winchester beneath the separator while the evaporated liquor or/

or "evaporate" collected in the winchester beneath the condensers.

The apparatus was allowed to run for a considerable period of time to allow the steady conditions to be reached.

Readings were then taken on "separate" and "evaporate" receivers and the following readings taken every 2 minutes for at least half an hour.

(1) Inlet temperature of the heating water

(2) Outlet temperature of the heating water

(3) Vacuum in cms. of mercury

(4) Pressure of the heating water

(5) Temperature of the feed in the bucket.

Readings for two runs of the travelling thermocouple, one going up the tube and the other going down, were taken. The mean of the two readings was taken.

Since a thermocouple, of the sort being used, has a time-lag of about 1 min., the thermocouple was allowed to remain for 1 minute in each position before the reading on the galvanometer scale was taken.

A maximum temperature was found at a certain position and the temperature taken at 1" intervals for 6" on either side of this position to ascertain the true maximum temperature position. The Section above this point was considered/

considered to be the boiling section and that below, the non-boiling section of the evaporator tube.

This procedure was repeated for different feed rates as shown in the table of results.

CALCULATION OF RESULTS.

The rate of feed was obtained by adding the amounts of "separate" and "evaporate", and knowing the duration of test, the rate of feed was calculated in lbs. per hour.

The heat load was found out as follows:

(a) The temperature of the steam and water at top tube plate was found out from a knowledge of the difference in temp. between the liquid there and the heating water in the jacket at the exit. The latent heat of steam at that temp. was found out from a steam-table. It was multiplied by the pounds of "evaporate" per hour which gave the total heat of steam at that temp.

(b) The total heat of water was found out by multiplying the "separate" per hour by the total heat of water at the temperature of water at top-tube plate.

(c) The total heat of water at the temperature of Bottom Tube Plate was found out by multiplying the total feed per hour by the total heat of water at feed temperature.

Then/

Then, Total heat load $Q = a + b - c$. The increase in the sensible heat (Q_{NB}) of the feed in being raised from the feed temperature to the maximum temperature of the liquid (obtained by multiplying the pounds of feed per hour by the difference in feed temperature and the maximum temperature of the liquid) subtracted from the total heat load gives the heat load in the boiling section, Q_B.

$$Q - Q_{NB} = Q_B$$

(d) Difference in temperature between the liquid in the evaporator tube and the heating water in the jacket was plotted against the distance from the bottom of the evaporator tube (Fig. 10). The area under the resulting curve divided by the length of the abscissa gave an integrated overall temperature drop, ΔT_m, based on the entire length of the tube. By considering the individual sections in the same way, the overall temperature drops in the non-boiling and boiling sections of the tube, ΔT_{NB} & ΔT_B, were obtained.
(e) Total heating area, A, of the tube was found out from the length and the mean of the inside and outside diameters of the tube. The non-boiling and the boiling areas, A_{NB} & A_B, were calculated from the corresponding non-boiling and boiling lengths of the tube. The overall heat transfer coefficient/

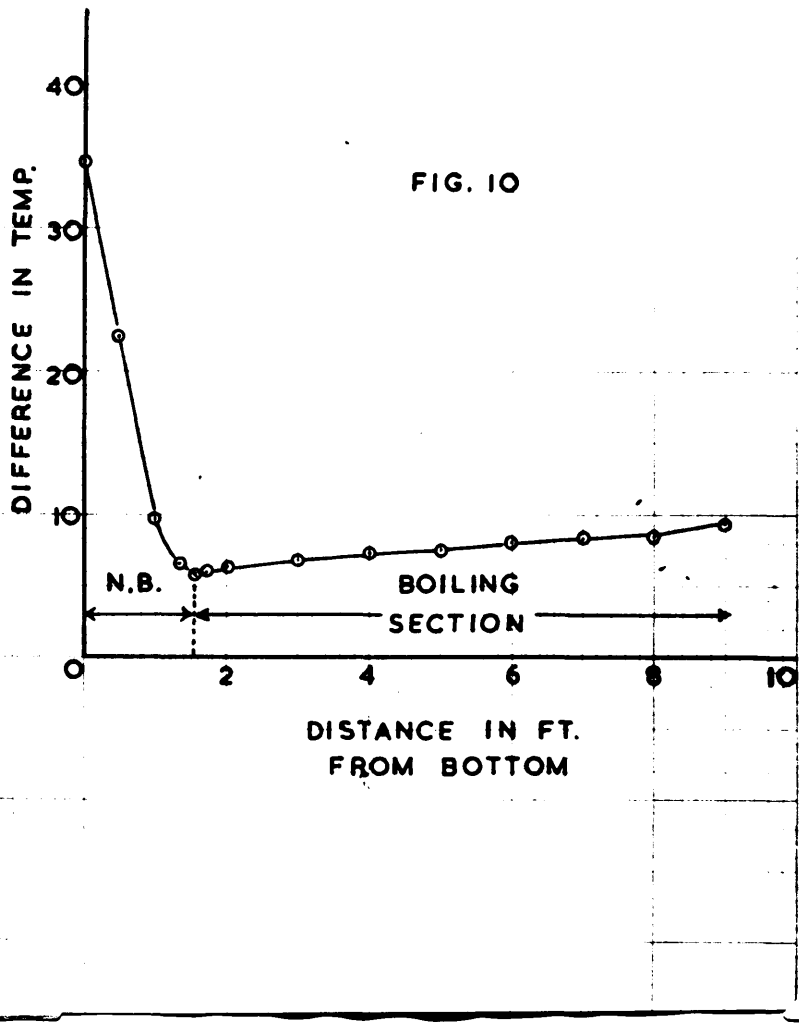

40
30
20
10
0
DIFFERENCE IN TEMP.
N.B.
BOILING
SECTION
2
4
6
8
10
DISTANCE IN FT.
FROM BOTTOM

FIG. 10

FIG. 10

coefficient, U, based on the entire length of the tube was calculated from

$$U = \frac{Q}{A, \Delta T_m}$$

and the overall heat transfer coefficient in the boiling section of the tube, U_B, was calculated from

$$U_B = \frac{Q_B}{A_B \, . \, \Delta T_B}$$

Then the film coefficient, h, based on the entire length of the tube was found out from the relation

$$\frac{1}{U} - \left(\frac{1}{H_A} + \frac{1}{H_W}\right) = \frac{1}{h}$$

and the boiling film coefficient, h_B, was found out from

$$\frac{1}{U_B} - \left(\frac{1}{H_A} + \frac{1}{H_W}\right) = \frac{1}{h_B}$$

The data are given in Table 3.

Comparison of the results with those of Rumford (54) done under comparable conditions show notable differences. There is a marked stability in the value of the overall heat transfer coefficients both when the boiling section of the evaporator tube and also the whole tube are considered, though the feed rate was varied by about 500%. Under comparable conditions/

TABLE - 3

Boiling Film Coefficient of Water in Steel Tube Evaporator

Heating Water at 80° C.

Vacuum - 65 cms. of Hg.

Test No.	Feed lbs/hr	Evap^{n}. lbs/hr	% Evap^{n},	Feed Temp. °F.	T.T.P. Temp. °F.	Max^{m}. Feed Temp. °F	Q B.Th.U./hr.	ΔT_m °F.	U B.Th.U./hr.	A_B sq.ft.	Q_B B.Th.U./hr.	ΔT_B °F	U_B B.Th.U./hr.	h_B B.Th.U./hr.
1.	21.39	12.12	56.6	57.2	153.8	161.8	14260	21.95	451	1.298	12022	16.92	548	1905
2.	25.84	11.63	45.0	75.2	151.1	160.3	13682	23,70	401	1.295	11470	19.27	460	1144
3.	33.19	11.33	34.2	59.0	153.8	162.0	14548	21.52	470	1.219	11128	16.92	540	1812
4.	33.98	11.18	32.9	56.3	152.1	161.4	14517	23.20	435	1.217	10965	18.78	480	1277
5.	39.32	10.36	26.4	65.3	154.4	163.2	13924	22.10	440	1.193	10074	16.16	523	1634
6.	39.69	10.39	26.2	56.3	154.7	162.1	14337	21.17	471	1.192	10142	16.54	515	1559
7.	44.89	10.49	23.2	57.2	153.1	162.1	14866	22.30	463	1.164	10160	15.77	554	1980
8.	45.25	10.00	22.1	51.8	153.7	163.4	14670	21.72	470	1.164	9620	16.20	510	1514
9.	47.71	10.46	21.9	66.2	154.8	164.1	14750	21.84	470	1.140	10080	15.55	569	2188
10.	51.01	9.88	19.4	54.0	153.5	162.1	15020	23.40	446	1.139	9510	16.35	511	1522
11.	54.71	9.71	17.7	54.5	154.7	163.2	15250	22.10	465	1.088	9300	15.42	555	1993
12.	65.70	9.55	14.5	55.4	153.9	162.1	16090	25.70	435	1.060	9080	16.36	524	1644
13.	68.90	9.75	14.2	58.1	156.0	164.1	16540	23.48	489	1.036	9240	14.63	610	2950
14.	86.63	8.83	10.2	52.7	155.8	164.3	17810	26.53	467	0.902	8145	15.18	595	2627
15.	100.15	8.75	8.7	54.5	156.2	164.1	19360	26.93	500	0.879	8380	14.91	640	3817

conditions of heat load and difference of temperatures in the boiling section of the evaporator an average constant value of overall heat transfer coefficient of 540 B.Th.U./Hr. was obtained for the boiling section whereas that of Rumford increased continually with the increased rate of feed. It is notable also that the boiling film heat transfer coefficients are fairly consistent and do not reach any unreal value with the increased rate of feed. According to the present results it seems that the conclusion arrived at by Rumford "that the artificial division of a vertical tube evaporator into non-boiling and boiling zones is not justifiable" is not sustained.

PART I

SECTION B

GLASS TUBE EVAPORATOR

PART I.

SECTION B.

GLASS TUBE EVAPORATOR.

To obtain a visual picture of the mechanism of the heat transfer to boiling liquids, a Pyrex glass tube vertical evaporator was erected. The experimental liquids were water, alcohol, toluene and an aqueous solution of a wetting agent.

APPARATUS:

The evaporator constructed is shown in Fig. (11). It consisted of an outer jacket 8 ft. 1 in. in length and 1 9/16 in. O.D. The inner glass tube consisted of $\frac{7}{8}$" o.d. & $\frac{3}{4}$" i.d. pyrex glass tube and 9 feet long with two quick-fit B 24 ground glass male joints at both the ends. The inner tube was heated for 7' $10\frac{1}{2}$" of its length by the hot water in the jacket giving a total heating area of 1.681 sq.ft. The outer jacket had two side tubes about $\frac{7}{8}$" o.d. and 3" long and fused at the bottom and top for the heating water inlet and outlet. The inner glass tube was fixed to the outer jacket by means of two rubber bungs which were then tied with wires and/

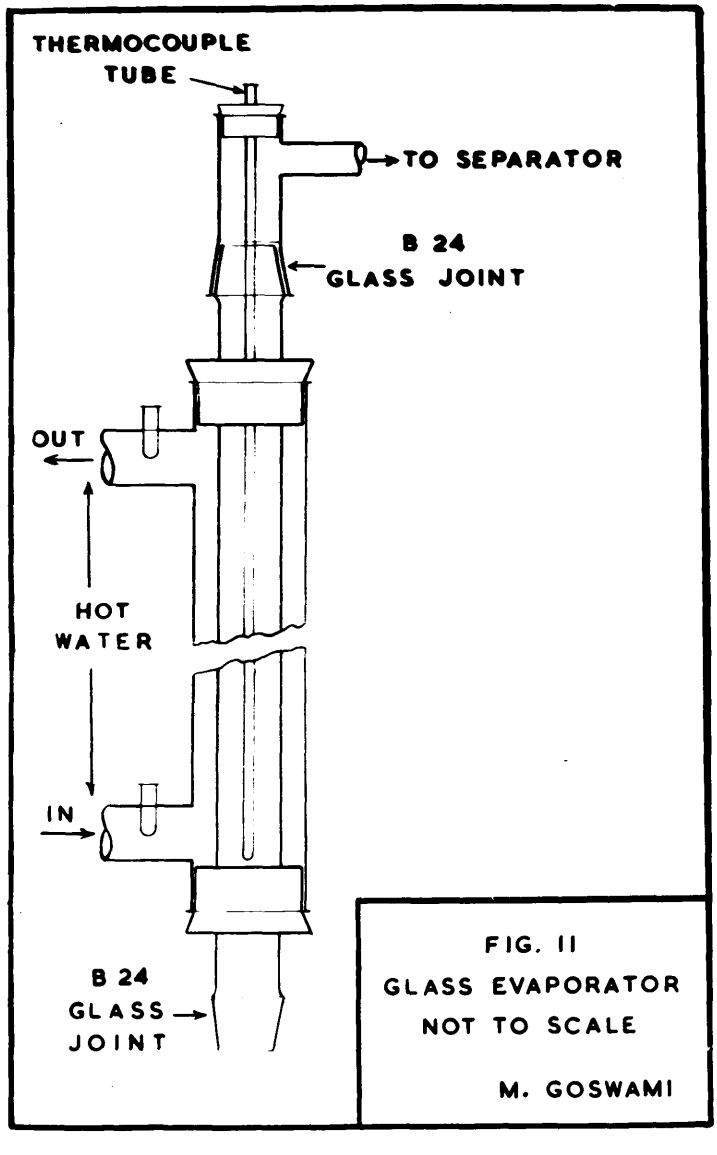

THERMOCOUPLE
TUBE
TO SEPARATOR
B 24
GLASS JOINT
OUT
HOT
WATER
IN
B 24
GLASS
JOINT
FIG. II
GLASS EVAPORATOR
NOT TO SCALE
M. GOSWAMI

FIG. II.

and ropes with the main jacket. A straight piece of glass tube ($\frac{7}{8}$" o.d. and 1' 3" long) with a side tube (about $\frac{1}{2}$" o.d.) was connected to the upper ground glass joint by means of a B 24 female ground glass joint. The side arm of this straight piece was joined by a rubber tube to the liquid-vapour separator and the condenser system of the previous apparatus, a 3/32" o.d. glass tube jacket for the travelling thermocouple was fixed from the top-piece of the evaporator tube by means of a rubber bung. The side arms of the outer jacket were connected to two Tee-pieces for registering the temperatures of the inlet and outlet heating water through thermopockets. The bottom Tee-piece was connected by a 1" copper hose from the hot water tank through a centrifugal pump. An iron pipe led the hot water from the top Tee-piece back to the heating tank.

A number of experiments were carried out with this unit. Unfortunately the thermocouple system gave much trouble and it is doubtful whether any of the results were quantitatively valuable. Nevertheless, the general picture of boiling formed showed that air bubbles exercised a preponderant effect on the whole phenomenon. The apparatus was therefore remodelled using a special deaerator.

DEAERATOR/

DEAERATOR.

There is much dissolved air in the feed liquid and the ordinary flowmeter gives much trouble through the leakage of air through the stop cock. This air finds its way inside the evaporator tube and give rise to considerable stirring of the liquid in the heating zone, thus disturbing the natural circulation. Hence some devices had to be adopted to get rid of the air inside the evaporator.

The ordinary flowmeter was modified. Both the flowmeters are shown in Fig. (12). The top of the modified flowmeter was connected with a glass tube to the vacuum line to a point just after the vacuum gauge (Fig. 6). As soon as the feed enters the stop cock, dissolved air gets separated forming bubbles. These bubbles after entering the air separator find their way out through the vacuum line while the air-free liquid is sucked in by the evaporator tube. The arrangement was quite satisfactory for the purpose for which it was constructed.

The thermocouples which had been the cause of failure were replaced by a new set of thermocouples made from wires of the same specifications. They were standardised and every now and then their performances were checked.

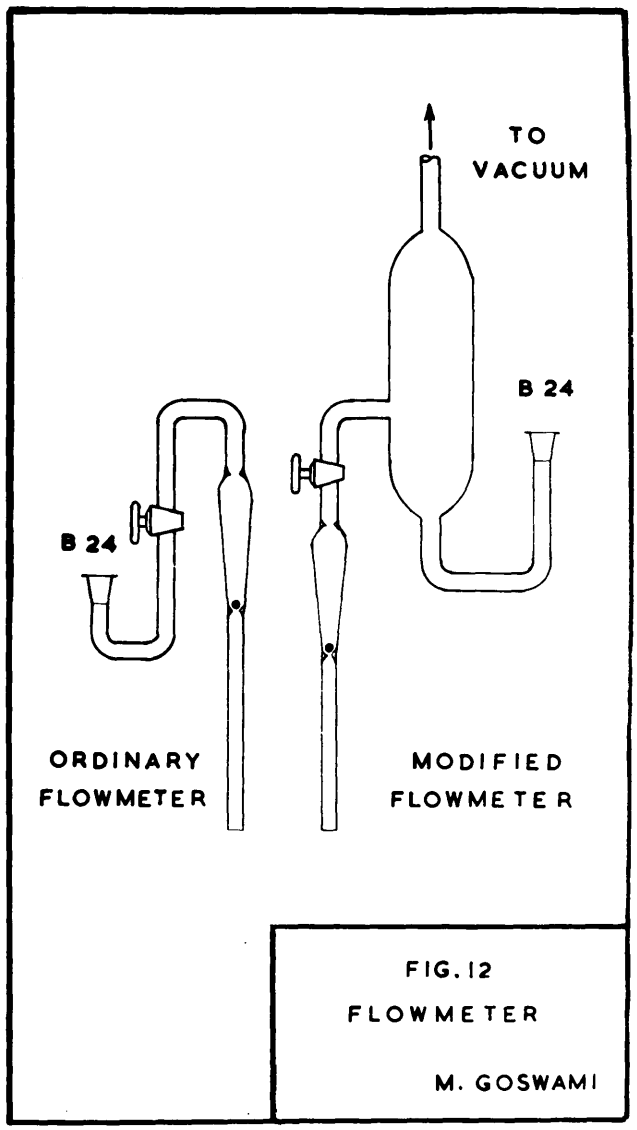

TO
VACUUM
B 24
B 24
ORDINARY
FLOWMETER
MODIFIED
FLOWMETER
FIG. 12
FLOWMETER
M. GOSWAMI

FIG. 12

PART I.

SECTION B.

(a) DETERMINATION OF OVERALL HEAT TRANSFER COEFFICIENT FOR WATER.

The test on the glass evaporator were carried out by the same method which have been described in the first section. The operating conditions under which the experiments were carried out are given at the top of the table of data. The data are given in Table 4.

It would have been possible to determine the annular film and wall coefficients for the glass evaporator in the same way as was done for the stainless steel unit. Foust & Thompson (26) measured the coefficient from water to water in a double-pipe heat exchanger, the inner pipes being of PYREX glass tubes, 1" and 2" in diameter. The non-boiling coefficients could be computed from Dittus-Boelter equation for the stream inside the Pyrex, but the equation gave low value for the annular coefficients. Evidence indicating the thermal conductivity of pyrex borosilicate glassware is higher than published values is discussed. Unfortunately no data are given.

However,/

TABLE - 4

Glass Tube Evaporator,

Feed - Cold Water (Deaerated)

Vacuum - 65 cms. of mercury,

Heating water at 80° C.

Test No.	Feed Rate lbs. /hr.	Evapn. lbs. /hr.	Evapn. %	Feed temp. °F.	ΔT_m °F	Q B.Th.U./hr.	U B.Th.U./hr.
1.	10.995	5.64	51.3	62.6	39.1	6532	100
2.	13.820	5.72	41.4	63.3	39.1	6835	104
3.	16.830	5.35	31.8	65.7	39.5	6660	101
4.	21.480	5.08	23.7	65.3	39.9	6743	101
5.	23.315	5.36	23.0	71.3	40.5	6977	103
6.	23.350	5.23	22.4	54.2	41.8	7260	103
7.	34.750	4.88	14.1	70.4	42.1	7288	103
8.	37.790	4.77	12.6	69.8	43.5	7405	102
9.	39.875	3.97	9.9	56.3	43.1	7343	102
10.	41.200	4.50	10.9	69.4	43.8	7430	101

However, in the present work as the main idea was to make a visual study of the mechanism of heat transfer in boiling liquids in long tube vertical evaporators, the combined value of the annular film and wall coefficient was not determined in the beginning. For that reason the boiling film heat transfer coefficient could not be calculated. Only overall heat transfer coefficients are given in the table.

From the table of data it will be seen that though the rate of feed has been varied by about 400%, there is a remarkable constancy in the value of the overall heat transfer coefficient, the average value being 102 B.Th.U./hr./ $ft^2/^oF$. This low value of the overall heat transfer coefficient, even though the difference of temperatures between the heating water and the water inside the evaporator tubes are fairly high, is mainly due to the low conductivity of the glass wall.

PART I.

SECTION B.

(b) EFFECT OF SURFACE TENSION.

The surface tension of the liquid, and the ability of the liquid to wet the heating surface, apparently have a strong influence on the heat transfer coefficient to boiling liquids.

Work was taken up to get a visual picture of the extent to which the mechanism of boiling in a long tube vertical evaporator is affected due to lowering the surface tension. A 0.1% aqueous solution of Perminal W.A. (an I.C.I. product) at ordinary room temperature was chosen as the feed liquid.

The experimental procedure and the calculation of the results were the same as before. Experimental conditions are given at the head of Table of Results. The results are given in Table 5.

The Deaeration device could not be used in this case because the stable bubbles formed at the top of the air-liquid separator had a tendency to shoot up to the vacuum line. The rubber tube connection between the Deaerator and the vacuum line was controlled by a screw-clip so that only some of the air alone could escape. The bulb of the Deaerator remained/

T A B L E - 5

Glass Tube Evaporator (Partial Deaeration)

Feed - 0.1 % Perminal W.A. soln. in water.

Vacuum - 65 cms. of mercury.

Heating Water at 80° C.

Test No.	Feed Rate lbs. /hr.	Evapn. lbs. /hr.	Evapn. %	Feed Temp. °F.	ΔT_m °F.	Q B.Th.U./hr.	U B.Th.U./hr.
1.	13.17	5.68	43.2	60.70	37.0	6804	109.5
2.	14.85	5.87	39.5	81.85	36.5	6818	111.5
3.	21.59	5.29	24.5	79.35	37.9	6678	105.0
4.	26.13	4.98	19.1	77.90	40.0	6681	98.5
5.	28.94	4.79	16.5	68.20	42.8	6955	97.0
6.	33.67	4.58	13.6	68.00	43.8	7087	96.5
7.	42.80	3.93	9.2	66.20	48.2	7033	87.0

remained full throughout and allowed the froth to separate so that only a clear liquid from the bottom of the Deaerator entered the evaporator tube.

It has already been mentioned that Jakob & Linke (34) under similar conditions of experiments obtained a 23% increase in the boiling heat transfer coefficient over that for water by lowering the surface tension by 45%. Insinger & Bliss (31) also got a 20% increase of the boiling heat transfer coefficient by reducing the surface tension by 27%.

Very striking results were obtained by Stroebe, Baker & Badger (59) on the heat transfer coefficient to boiling liquids, in a long tube vertical evaporator, due to reduced surface tension. They found that the increase in the coefficient due to a lowering of the surface tension was much greater than had previously been observed or generally recognised. A small amount of "Duponol", a soapless detergent, when added to the feed water lowering the surface tension by almost 50%, the boiling film coefficients obtained were 2 to 4 times higher than with pure water under the same conditions.

These suggest that the surface tension has a definite influence on the mechanism of boiling and should be included in any correlation of boiling film coefficients.

From/

From a comparison of the data on water and that on the Perminal solution it will be seen that approximately there has been no appreciable change in the value of the overall heat transfer coefficient. It should be noted here that in the present case the overall heat transfer coefficient was calculated considering the entire length of the tube as opposed to the boiling film coefficient used by the other investigators. Another point of importance is that both the set of experiments were not carried under similar conditions, because the feed water was completely free from dissolved air, the Perminal solution was not. In the circumstances no comments can be made of the effect of lowering the surface tension on the heat transfer coefficient. Anyway, a distinct effect on the nature of boiling due to the addition of Perminal W.A. was noticed and it will be discussed in the proper place.

PART I.

SECTION B.

(c) DETERMINATION OF OVERALL HEAT TRANSFER COEFFICIENT FOR ALCOHOL IN THE GLASS TUBE EVAPORATOR:

EXPERIMENTAL PROCEDURE.

The apparatus was dismantled, thoroughly washed with distilled water and then with absolute alcohol, dried and reassembled.

The same experimental procedure was followed as before. As the boiling point of absolute alcohol is much lower than that of water, the heating water in the jacket was circulated at a temperature of 50° C. (122°F) and a lower vacuum was maintained. The experimental conditions are listed at the head of the table of data.

CALCULATIONS.

The method of calculation was the same as used before. The data are given in Table 6.

Though the feed rate is varied by 500%, there is a remarkable constancy in the value of the overall heat transfer coefficient.

Some/

T A B L E - 6

Glass Tube Evaporator.

Feed - Cold Alcohol (Deaerated)

Vacuum - 61.8 cms. of mercury.

Heating Water at 80° C.

Test No.	Feed Rate lbs. /hr.	Evapn. lbs. /hr.	Evapn. %	Feed Temp. °F.	ΔT_m °F.	Q B.Th.U./hr.	U B.Th.U./hr.
1.	6.09	3.216	52.8	72.9	14.16	1355	57.0
2.	6.47	3.112	48.1	73.0	14.69	1321	53.5
3.	7.75	3.145	40.6	71.6	14.89	1367	54.5
4.	7.78	2.972	38.2	71.3	15.48	1300	50.0
5.	8.31	3.288	39.6	69.8	14.40	1438	59.5
6.	8.44	3.250	38.5	65.9	14.60	1453	59.0
7.	11.11	3.008	27.1	66.2	15.10	1422	56.0
8.	13.36	2.770	20.7	64.4	17.46	1400	47.5
9.	13.91	3.007	21.6	69.8	15.70	1460	55.5
10.	31.62	2.446	7.7	64.0	19.57	1749	53.0

Some distinctive phenomenon in boiling was observed when no air was present in the feed liquid. Boiling was not taking place at a point where the maximum temperature was found. Irrespective of the rate of feed the boiling was confined to the upper part of the evaporator tube, bubbles forming only at an irregular surface of the tube. This will be discussed in detail in the general discussion on the nature of boiling observed.

PART I.

SECTION B.

(d) DETERMINATION OF OVERALL HEAT TRANSFER COEFFICIENT FOR TOLUENE IN GLASS TUBE EVAPORATOR:

EXPERIMENTAL PROCEDURE:

The apparatus was dismantled, thoroughly washed and dried and set up again.

The experimental procedure followed was the same as in the case of absolute alcohol.

Since toluene has a low specific heat and low latent heat of vapourisation, the rate of evaporation is very fast. Consequently, if the feed rate is very slow, all the feed escapes from the evaporator tube as vapour leaving the upper part of the evaporator tube walls dry. So in order to keep a liquid film in the upper part of the evaporator tube, the feed rate had to be maintained fairly high. At the same time very high rates of feed were not possible for some technical difficulty, such as the amount of toluene available.

CALCULATIONS:

The difficulty in calculation in this case was that the latent/

latent heat of vapourisation of toluene at different temperatures were not available anywhere in the literature. The equation of Clapeyron-Clausius was used for this purpose:

$$\frac{dP}{dT} = \frac{L}{T(V_G - V_L)}$$

where

L = Molal latent heat,

T = Temperature, ^oA.

P = Vapour pressure, mm of Hg.,

V_G = Specific volume of vapour,

V_L = Specific volume of liquid,

Neglecting V_L in comparison with V_G and assuming the vapour obeys gas Law ~~whence~~

$$\frac{dP}{dT} = \frac{LP}{RT^2}$$

Whence,

$$L = 2 \times 2.303 \frac{T_1 \ T_2}{T_2 - T_1} \log_{10} \frac{P_2}{P_1}$$

for a very small range of temperature over which L can be safely assumed to be constant.

Now, if log P is plotted against 1/T, then an empirical relationship is obtained. From this curve two very close lying values of temperature can be chosen and the values of

log/

log P corresponding to them can be read from the curve which for all practical purposes is a straight line. Thus for two temperatures T_1 and T_2 the corresponding values of log P_1 and log P_2 can be found out. Then, by substituting these values in the above relationship, the value of the latent heat can be found out.

Now, in these experiments the average temperature at the exit was 68° C. or 341° A.

Therefore, $1/T = 29.35 \times 10^{-4}$

From the graph (Fig. 13)

when $\log_{10} P_1 = 2.26$, $1/T_1 = 29.4 \times 10^{-4}$

$\log_{10} P_2 = 2.28$, $1/T_2 = 29.3 \times 10^{-4}$

therefore $T_1 = 340.3$

and $T_2 = 341.5$

Hence

$$L = \frac{2 \times 2.303 \ (T_1 \ T_2)}{92 \ (T_2 - T_1)} \log_{10} \frac{P_2}{P_1} \text{ cal. gm.}$$

(Mol. wt. of toluene = 92)

$$= \frac{2 \times 2.303 \times 341.5 \times 340.3}{92 \times 1.2} \times 0.02 \times 1.8 \text{ B.Th.U./lb.}$$

$= 174.8$ B.Th.U./lb.

Specific/

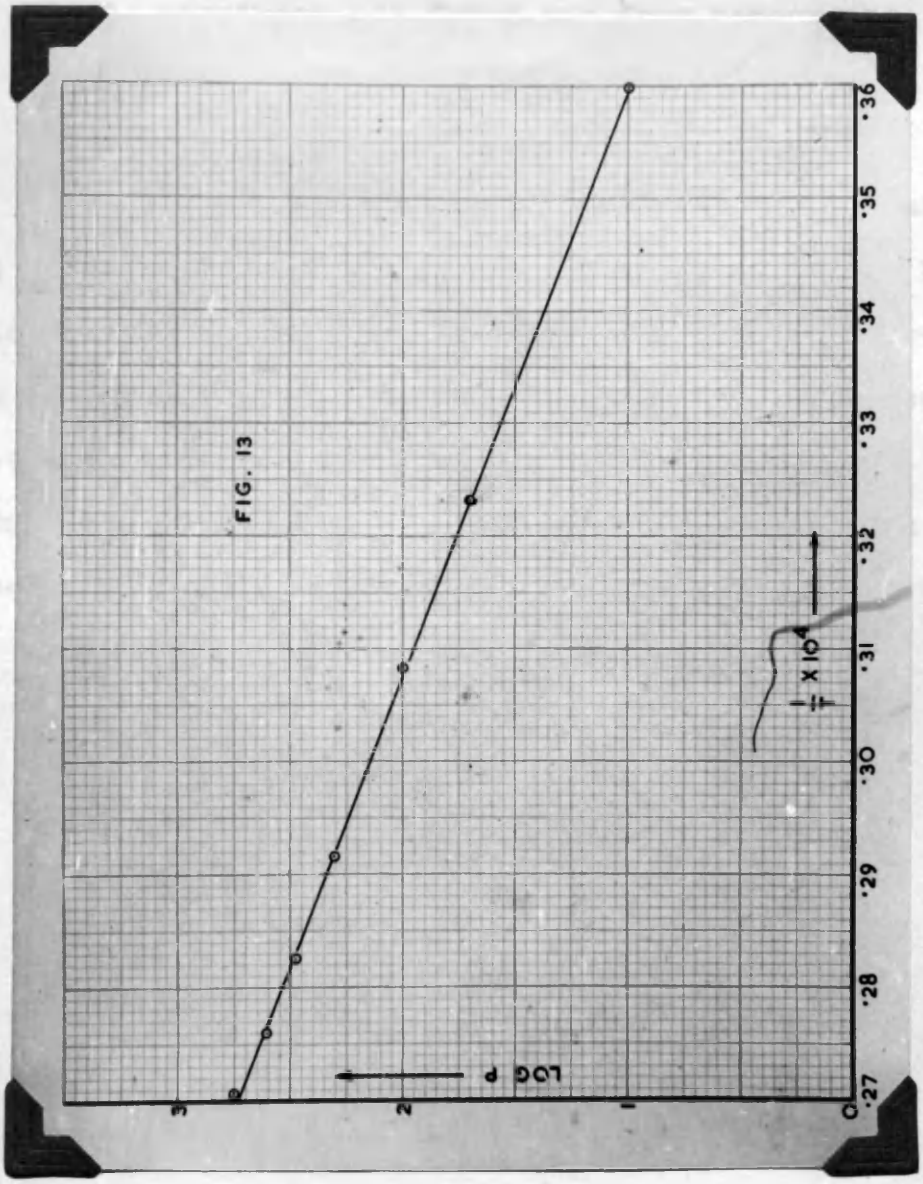


FIG. 13

Specific heat of toluene at the average of initial and exit temperatures was found out from nomographs.

Rest of the calculations were done as before.

Results are given in Table 7.

It will be noticed that the deaerator was not used in this case because there would not take place of any boiling of toluene if it was completely free from air. Though the feed rate was varied by more than 200%, there is a remarkable constancy in the value of the overall heat transfer coefficient. A remarkable feature here is the high percentage evaporation of the feed liquid.

TABLE - 7

Glass Tube Evaporator

Feed - Cold Toluene (Without Deaerator)

Vacuum - 65 cms. of mercury.

Heating Water at 80° C.

Test No.	Feed Rate lbs. /hr.	Evapn. lbs. /hr.	Evapn. %	Feed Temp. °F.	ΔT_m °F.	Q B.Th.U./hr.	U B.Th.U./hr.
1.	28.88	13.60	47.1	68.0	24.07	3456	85.5
2.	31.06	13.74	44.2	68.0	24.45	3563	87.0
3.	31.64	13.85	43.8	68.5	25.53	3584	83.5
4.	34.29	14.00	40.8	68.4	25.10	3725	88.5
5.	34.48	14.81	43.0	70.5	27.50	3803	82.5
6.	35.51	13.39	37.7	68.7	26.42	3662	82.5
7.	41.88	13.84	33.4	66.2	26.80	4037	89.5
8.	44.10	13.50	30.6	69.3	27.25	4108	88.0
9.	59.38	14.83	25.0	66.2	32.05	4820	89.0

PART I.

SECTION B.

(e) EXTENSION OF WORK ON WATER AND ALCOHOL.

The main aim of this work was to compare the values of the heat transfer coefficient for water, alcohol and toluene for a range of difference of temperatures. The feed rate and the pressure inside the evaporator tube in both the cases of water and alcohol had to be so adjusted that same range of difference of temperatures were obtained. This was achieved in a very simple way by adjusting the pressure and the rate of feed so that the boiling always commenced at two feet from the bottom of the evaporator tube.

The overall heat transfer coefficients in the boiling and the non-boiling sections also have been calculated in these cases considering the amount of heat transferred in each section separately.

CALCULATIONS:

(a) The total amount of heat transferred in the whole evaporator tube is

Q = (Heat required to raise the temperature of the water at the entrance to that at the exit) + (Heat in the vapour at the temperature at exit)

(b)/

(b) The amount of heat transferred in the non-boiling section is

Q_{NB} = Heat required to bring the temperature of the feed at the entrance to that at the maximum temperature in the tube

(c) The amount of heat transferred in the boiling section is

$$Q_B = Q - Q_{NB}$$

The difference of temperatures in the non-boiling and the boiling sections were obtained by the graphical integration of the curve considering the different sections concerned.

As the boiling always took place at two feet from the bottom of the evaporator tube, the following were fixed in all cases:

Heating area = 0.427 sq.ft.

Boiling area = 1.254 sq.ft.

The data are given in Tables 8 & 9. The experimental conditions are also listed in the tables.

Cessna, Lientz and Badger (15) found that for constant boiling lengths, the boiling film coefficients increase with decreasing Delta-T, a fact, which they admit, is contrary to observations/

TABLE - 8

GLASS TUBE EVAPORATOR WITH DEAERATOR

Feed - Cold water

Heating water at 80° C.

Test No.	Feed Rate lbs /hr.	Evapn. lbs. /hr.	Evapn. %	Feed Temp. °F.	T,T.P. Temp. °F.	Maxm. Liquid Temp. °F.	Vacuum cms. Hg.	Q B.Th.U./hr.	ΔT_m °F	U B.Th.U/hr.	Q_{NB} B.Th.U/hr.	ΔT_{NB} °F	U_{NB} B.Th.U/hr.	Q_B B.Th.U./hr	ΔT_B °F	U_B B.Th.U./hr.
1.	11.74	1.324	11.3	62.1	164.9	170.9	50.0	2532	15.42	98.0	1276	31.42	96.0	1256	9.53	105.0
2.	11.72	1.368	11.7	62.3	164.4	169.9	50.0	2567	16.00	95.5	1260	31.70	93.0	1307	10.05	104.0
3.	12.91	1.602	12.4	70.7	162.4	167.7	51.2	2789	17.41	95.5	1252	36.37	81.0	1537	11.88	103.5
4.	16.75	2.315	13.8	67.1	157.2	163.5	55.0	3835	25.17	90.5	1614	49.15	77.0	2221	16.82	105.5
5.	16.45	2.406	14.6	63.0	157.2	161.8	55.0	3968	25.30	93.5	1625	49.15	77.5	2343	16.88	111.0
6.	18.48	3.004	16.3	66.4	152.5	158.0	57.4	4617	29.48	93.5	1694	53.30	74.5	2923	21.73	107.5
7.	18.63	2.950	15.8	66.6	152.7	157.8	57.4	4795	29.96	95.5	1699	54.00	74.0	3060	21.58	113.0
8.	18.77	3.345	17.8	63.0	149.6	155.8	59.0	4999	32.35	92.0	1742	55.55	73.5	3257	24.12	108.0
9.	20.50	3.675	17.9	68.4	148.6	154.4	60.3	5354	32.68	97.5	1764	54.75	75.5	3590	24.10	119.0
10.	21.12	3.675	17.4	66.2	148.5	154.6	60.3	5448	33.65	96.5	1866	57.10	76.5	3582	25.36	113.0
11.	26.92	4.713	17.5	65.0	142.5	149.4	65.0	6866	38.93	105.0	2265	61.58	86.0	4601	30.63	120.0
12.	25.81	4.660	18.6	64.4	142.5	149.2	65.0	6735	39.10	102.5	2265	61.20	87.0	4470	31.62	113.0

TABLE - 9

Glass Tube Evaporator with Deaerator.

Feed - Cold Alcohol.

Heating Water at 50°C. in Tests 1 to 5

at 80°C. in Tests 6 to 15.

Test No.	Feed Rate lbs. /hr.	Evapn. lbs. /hr.	Evapn. %	Feed Temp. °F.	T.T.P. Temp. °F.	Maxm. Liquid Temp. °F.	Vacuum cms. Hg.	Q B.Th.U./hr.	ΔT_m °F	U B.Th.U./hr.	Q_{NB} B.Th.U./hr.	ΔT_{NB} °F	U_{NB} B.Th.U./hr.	Q_B B.Th.U./hr.	ΔT_B °F	U_B B.Th.U./hr.
1.	17.08	4.46	26.1	61.9	102.9	109.4	63.7	2113	20.45	61.5	490	31.15	37.0	1625	16.85	77.0
2.	18.57	4.45	23.9	63.5	102.2	108.7	64.3	2121	21.10	60.0	495	30.60	38.0	1626	17.65	73.5
3.	16.67	4.67	28.1	60.3	101.9	108.0	64.3	2185	20.70	60.0	477	32.22	34.5	1708	17.90	76.0
4.	20.54	4.97	24.2	68.7	100.6	107.4	66.2	2282	22.20	61.0	485	30.80	37.0	1797	18.90	76.0
5.	19.06	4.95	26.0	66.8	100.6	106.0	66.2	2262	22.82	59.0	455	32.04	33.5	1807	19.32	74.5
6.	20.10	5.06	25.1	83.9	156.2	158.3	25.7	2746	25.15	65.0	1016	45.55	52.5	1730	18.75	74.0
7.	20.31	5.25	25.9	77.4	156.6	159.2	24.7	3011	26.53	67.5	1118	51.20	51.0	1893	18.20	83.0
8.	18.35	5.15	28.2	64.9	157.1	158.3	25.4	3014	27.10	66.5	1130	52.75	50.0	1884	18.45	81.5
9.	19.78	5.15	26.1	71.6	154.2	158.0	26.4	2980	28.60	62.0	1142	52.85	50.5	1838	20.36	72.0
10.	19.04	5.46	28.7	76.2	154.0	157.3	28.0	3018	28.77	62.5	1035	50.98	47.5	1984	20.80	76.0
11.	21.35	6.65	31.2	78.8	151.5	154.8	32.0	3497	30.56	68.0	1087	51.86	49.0	2410	23.20	83.0
12.	18.77	6.83	36.4	65.3	150.8	153.6	32.0	3568	31.95	66.5	1087	53.80	47.5	2481	23.85	83.0
13.	24.55	8.26	33.7	73.4	145.2	148.7	37.6	4228	36.90	68.0	1220	58.70	48.5	3008	29.45	81.5
14.	19.65	8.27	42.1	65.3	146.0	150.1	36.6	4030	37.02	65.0	1083	58.70	44.0	2947	28.60	82.5
15.	29.46	7.75	26.3	71.6	146.7	148.8	37.2	4327	37.02	69.5	1494	61.40	57.0	2833	27.95	81.0

observations of other natural circulation evaporators.

In the present work, if log U_B is plotted against the difference in overall temperature drop in the boiling section (Fig. 14) it will be seen that there is a slight increase in the value of overall heat transfer coefficient with the increase of Delta-T. In the experiments of Cessna et al, the heat flux in the boiling section ranged between 30,000 and 200,000 B.Th.U. per hour. The boiling length remaining constant, at higher values of Delta-T the amount of vapour evolved will be greater and as shown by Partridge (48) a larger portion of the heating surface will be dry which might be the possible reason of the lower heat transfer coefficient observed by Cessna et al.

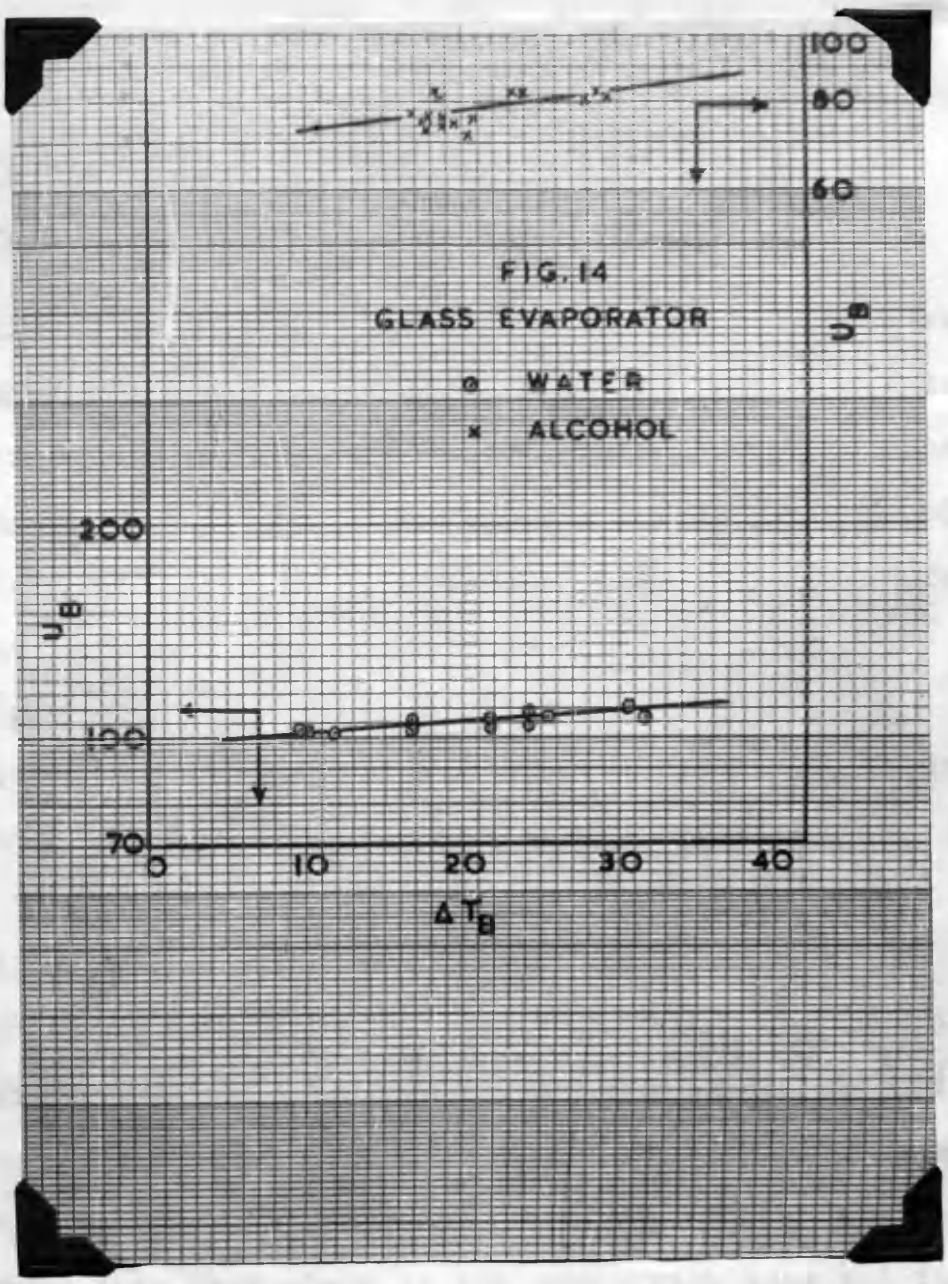


FIG. 14
GLASS EVAPORATOR
WATER
ALCOHOL
100
80
60
200
100
70
0
10
20
30
40
ΔT_B

FIG. 14

PART I.

SECTION B_1.

OBSERVATIONS ON THE NATURE OF BOILING IN A GLASS TUBE EVAPORATOR.

Kirschbaum and his co-workers (38a) first put forward the suggestion that the evaporator tube should be divided into two sections: (a) the heating section, and (b) the boiling section. In their apparatus a special glass plate was installed over the outlet of the evaporator tube, for the observation of boiling.

Brooks & Badger (13) were the first to present a correlation of the overall heat transfer coefficients in the boiling section of the evaporator tube. They erected a glass tube evaporator for the observation of the mechanism of boiling in such an externally heated tube.

Stroebe, Baker and Badger (59) observed the mechanisms of different types of action taking place through sight glasses suitably placed in the vapour head of the evaporator.

Badger (3) gave a qualitative discussion of the major phenomena that take place in the evaporator tube and the different types of boiling that may occur.

From/

From his experimental data Rumford (54) did not advocate the artificial division of a vertical tube evaporator into boiling and non-boiling zones. It was further stated that boiling occurred, at least in the film, throughout the length of the evaporator tube.

In the evaporator it was clear that a portion of the evaporator tube acted as a non-boiling section and heated the liquid up to the boiling point. The rest of the part of the evaporator tube acted as a boiling section where the actual evaporation took place. As can be expected, the non-boiling section of the evaporator tube increases in length with the increased feed rate. Up to the point where the actual boiling starts, the liquid is a steady rising column, and up to the point where the liquid has reached the maximum temperature, the hydrostatic and the friction heads are sufficient to prevent vapourisation of the liquid. Small bubbles of vapour are seen to be formed near this point and due to the superheated condition of the liquid the bubbles increase in size. As the liquid rises up further, this process of bubbles formation is further helped due to the pressure drop and there is a large increase in volume. The sensible heat is tranformed to latent heat in this region. Due to these two reasons there is a gradual drop in temperature of/

of the liquid as it travels up the evaporator tube. As the bubbles of vapour go higher up, they increase in size up to the full tube diameter and due to the high velocity of the vapour thin layers of liquid move upwards alongside the wall of the evaporator tube. But as the liquid is not in a continuous steady column, it tends to fall down for re-entrainment by fresh bubbles of vapour from below. There is thus an upward and downward movement of the liquid layer beginning from the point where it begins to boil.

There is dissolved air in the feed liquid and as soon as the feed liquid enters the flowmeter, small bubbles of air are formed which grow in size, as they go higher up, due to the release in pressure, and ultimately grow very rapidly at the point where boiling starts. The effect of this air is two-fold. Firstly, it agitates the liquid in the heating section and perhaps increases the heat transfer coefficient in that zone, and secondly it does not allow the liquid to take a good deal of superheat.

The upward and downward movement, as mentioned above, is by no means steady. It occurs in surges, the vigour of which is fluctuating.

Barbet (6) describes in great detail the appearance of the liquid when boiling occurs. He states that with low inlet velocities of the order of 0.3 ft. per sec., bubbles form, which coalesce, pushing slugs of liquid ahead of them. Due to the attrition at the walls of the tube, these slugs become thinner and finally break, allowing the vapour to pass through. If the vapourisation is sufficient, the liquid on the walls of the tube will move upward.

Barbet's results were verified by Brooks & Badger in a steam-jacketted glass tube evaporator and progressing up the tube the following types of action were encountered:

(1) A very short length in which the vapour and liquid more or less uniformly distributed both transversely and longitudinally in the tube.

(2) A section of "slug" action in which the transverse distribution is uniform, but the longitudinal is not, i.e. there are alternate slugs of liquid and vapour.

(3) A section in which the "film" action occurs, or the longitudinal distribution is uniform, but the transverse is not, i.e. there is a core of vapour with the liquid in a film of varying thickness on the walls of the tube. Annular rings mentioned by Barbet also appeared in this section.

They also observed that as the temperature difference and boiling/

boiling length increased two types of spray, "coarse" and "fine", began to appear. The coarse spray seemed to be travelling at a much lower velocity than the fine spray. In addition, the fine spray seemed to be in the centre of the tube, while the coarse was on the outside. Furthermore, the coarse appeared, in most cases, to be blown outward while the fine spray travelled up to the deflector in straight lines.

It will be seen that the phenomena as has already been described correspond to the phenomena (1) and (2) stated by Brooks and Badger. A Photograph of this slug action is shown Fig. (15).

Higher up in the evaporator tube the vapour moves upwards rapidly dragging the liquid still due to the high velocity of the vapour. The liquid moved alongside the tube in the form of rings. Another set of downcoming stream of rings meet this stream of rising rings. The densities of these rings vary in the longitudinal direction of the tube. Somewhere it is more dense whilst in the other it is less. The to and fro movement in the vertical direction goes on until a sudden gust of vapour carries the rings right up to the top and some find their way through the side arm into the separator. But if the temperature difference is high enough/

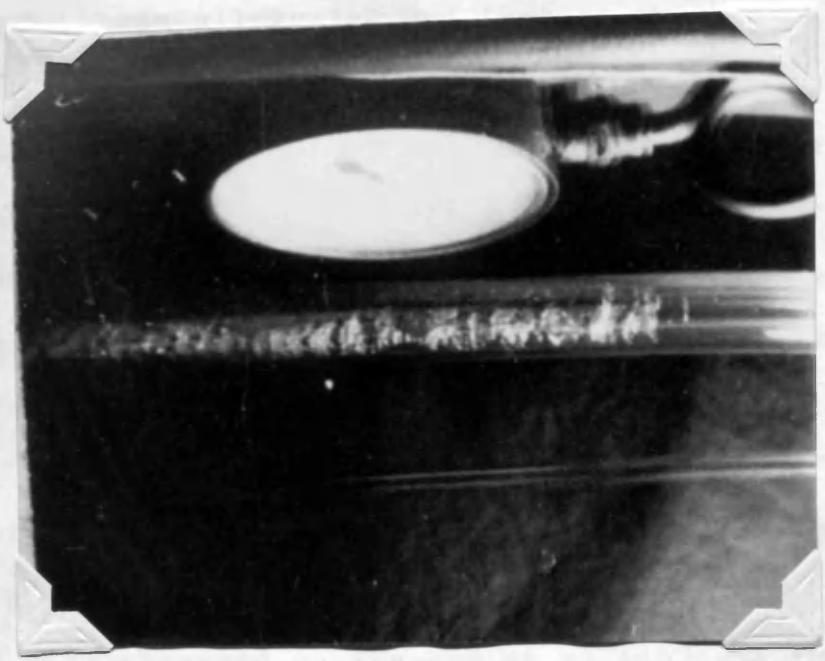


FIG. 16

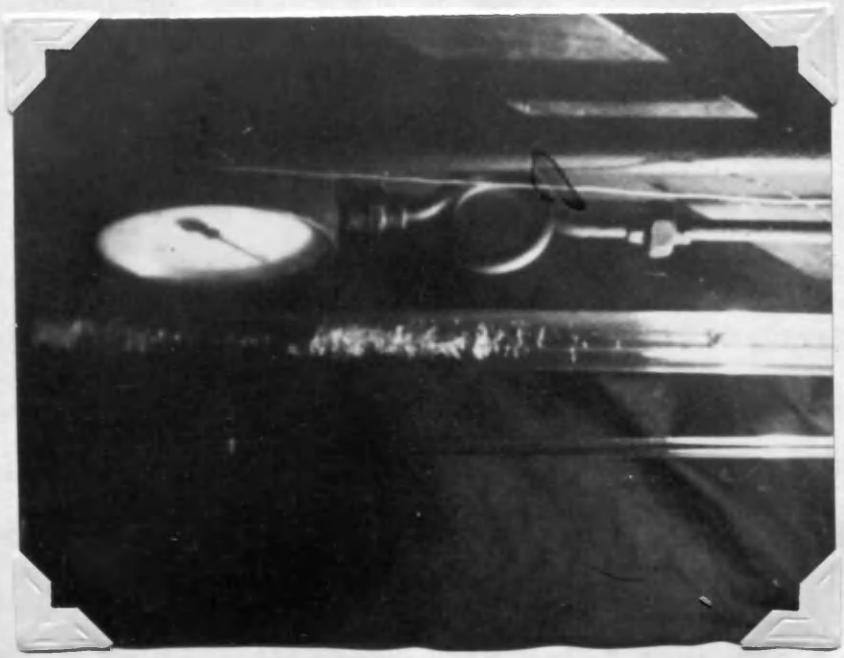


FIG. 15

enough, these all vanish and instead there is a film of liquid next to the tube wall and a core of vapour in the middle. When the velocity of feed is very low and the boiling length is great, there is still greater amount of vapour and sometimes the top of the tube becomes dry. This phenomena can be compared with the phenomenon (3) described by Brooks and Badger.

As regards the formation of "coarse" and "fine" spray nothing could be observed as there was no glass window in the separator. All the observations mentioned above have also been observed by Stroebe et al (59) and they give a photographic illustration of the liquid leaving the top of the evaporator tube.

EFFECT OF DEAERATION ON THE NATURE OF BOILING.

When the feed was free from air, the liquid rose in the evaporator tube in a steady column. It did not boil at a point where it should have, in accordance with the rate of feed. In the case of alcohol it was found that irrespective of the rate of feed, the column of liquid rose steadily until it reached an irregular surface (where the two glass tubes have been joined together) where bubbles of vapour began to form and it is only above this point that the boiling was visible/

visible. In the case of toluene no boiling took place at all if air was absent and in many cases the upper part of the tube was dry.

At very high feed rate, when boiling took place about 3 to 4 ft. above the bottom tube plate, gradual boiling occurred and maximum agitation set in within a few inches of the region of commencement. Here, there was no sharp demarcation of boiling and non-boiling sections. The point of minimum difference of temperature lay about the point where boiling commenced. The general behaviour of the fluid in the boiling section was same as before.

At low rates of feed, the range of the non-boiling section should be very small but instead of that the liquid rises higher up and boiling is visible only at the point of irregular surface as mentioned already. The travelling thermo-couple was moved at different heights in the evaporator tube in order to explore the point of highest temperature and that point was found to be much lower than the irregular surface where boiling was visible. This suggests that a point of minimum difference in temperature is not always an indication of change of state. Obviously much superheating of the liquid had taken place. Above the irregular surface though boiling was taking place, it was very smooth and confined to a small part of the tube/

tube. The upper part of the tube was quite dry and no liquid was being carried away to the separator. This was especially observed in the case of toluene as feed. If some air bubbles found their way in the feed, the whole system was tremendously disturbed and boiling took place at a much lower point and flashes of liquid and vapour were carried to the separator.

It will be remembered that in the discussion of the mechanism of nuclear boiling, the effect of dissolved air in the operation has already been mentioned. From the discussion in the previous paragraph it would appear that air has a great effect on the nature of boiling in such an evaporator. To what extent this does affect the heat transfer coefficient has so far not been investigated. In the non-boiling section aeration might increase the heat transfer due to agitation but in the upper part of the tube some blanketting effect might be expected. Industrial water is saturated with air and it is surprising that it has not been so far investigated in what way it affects heat transfer coefficients to boiling liquid.

At first sight, it would appear that at higher temperature drops the larger volumes of vapour and higher vapour velocities should increase fluid velocity and, therefore, give higher coefficients/

coefficients. A possible explanation for the reverse condition is found in a paper by Partridge (48) on the mechanism of formation of boiler scale. He showed, by examining the surface on which the steam bubbles were forming, that the area under the bubble is dry. Thus as the difference of temperature of the liquid film is increased and more bubbles are evolved, it is reasonable to assume that a larger portion of the surface will be covered with vapour and less will be wetted by the liquid and this is sufficient to account for the decrease in heat transfer coefficient under such conditions. This fact has been recognised by Stroebe et al (59) and in their correlation temperature difference is represented by a negative exponent. In the present glass tube evaporator it has been found that when the tube was dry at the top, specially in experiments with alcohol and toluene, the heat transfer coefficient fell considerably. It is worthwhile to mention here that in the copper tube evaporator, (next section) when the rate of feed is low and the boiling length of the tube is a large fraction of the whole the heat transfer coefficient is considerably lower and this effect is specially notable in the case of alcohol and toluene. The above might be the possible explanation. It is worth mentioning here that no formation of/

of any sort of vapour bubble in the film in the non-boiling section of the tube was found.

EFFECT OF LOWERING SURFACE TENSION:

The use of Perminal W.A. to produce a frothing liquid introduced some experimental difficulty. The deaeration device could not be used in the same way as used previously because of the tendency of the stable bubbles formed at the top of the air-liquid separator to shoot up the vacuum line (Fig.6). The rubber tube connection between the deaerator and the vacuum line had to be controlled by a screw-clip so that only a slight passage was there to let the air escape. The bulb of the deaerator remained full throughout the experiment and allowed the froth in the feed to separate so that only the clear liquid from the bottom entered the evaporator tube. Under these conditions small air bubbles arose from the bottom of the evaporator tube, and they increased in number as the liquid rose up the tube.

As the liquid was rising up the tube, it became heated and at a point boiling set in. The separation of the boiling and the non-boiling sections was quite distinct. The maximum temperature of the liquid was very near the line of demarcation between the boiling and the non-boiling sections.

On/

On adding wetting agent, the bubbles of vapour formed were very stable and as they rose they had no tendency to break down letting the liquid fall down along the evaporator tubewall. There was no "slug" action, i.e. there were no alternate slugs of liquid and vapour in the longitudinal direction. There was almost a continuous column of bubbles in the whole of the boiling section of the evaporator tube. A photographic illustration of this type of boiling is given (Fig. 16).

That the surface tension is an important factor in heat transfer to boiling liquids, both as affecting the size of the bubbles and the wetability of the surface, has been shown theoretically and experimentally by Jakob and his co-workers. It has been shown by Fritz (28) and Fritz & Ende (29) that a decrease in surface tension reduces the maximum size of the vapour bubbles, and also that the heat transfer coefficient from a liquid to a bubble of vapour is very large upon origin of the bubble, and decreases as the size of the bubble increases. It then follows that with a decrease in surface tension, the size of the bubbles evolved will be smaller, and consequently a greater average heat transfer coefficient can be expected.

As to the size of the bubbles formed with perminal solution/

solution, nothing striking was found. This might be due to the interference by the dissolved air which could not be altogether got rid of. Nothing remarkable was found in the change of heat transfer coefficient as well.

P A R T I

S E C T I O N C

COPPER TUBE EVAPORATOR

PART I.

SECTION C.

COPPER TUBE EVAPORATOR.

OBJECT:

Different metals have different conductivities and it is natural to think that surfaces of different metals will behave differently so far as the heat transfer coefficient is concerned. Many metal surfaces have been used by different investigators either in the plate or wire forms in finding out the film coefficients of boiling liquids. There is a disagreement concerning the relative order of merit of the surfaces of different metals from the point of view of heat transfer (7). The experiments of Sauer, and Cooper (55) showed that iron surfaces are better than copper, but Pridgeon and Badger (50) found the reverse to be true. The argument for the better performance of iron surface was that it gave more nuclei for boiling to start than the copper surface and hence compensated for the lower conductivity of the metal. It is also desirable that in the determination of film heat transfer coefficient, the whole of the resistance to heat transfer shall be centred in the film being investigated.

PART I.

SECTION C.

(a) DETERMINATION OF THE COMBINED ANNULAR FILM AND WALL COEFFICIENTS.

A copper tube was fitted in place of the stainless steel tube. The copper tube had the dimensions

Heating Length	=	9 ft. 2 in.
Internal diameter	=	15/32 in.
External diameter	=	19/32 in.

which gives

Cross section	=	0.0012 sq. ft.
Heating area	=	1.275 sq. ft.

The outside of the tube was cleaned with a piece of cloth and alcohol in order to get rid of any grease or dirt etc. In the first instance it was necessary to find out the value of the combined annular film and wall coefficients

$$\frac{1}{H_A} + \frac{1}{H_W}$$

for this copper tube, both at 80° C and 50° C. of heating water/

water, in order to calculate the liquid film coefficient. The apparatus was arranged in the same way as was used in the case of stainless steel tube (Fig.3). The experiments were carried out in the same way with different feed rates through the inner tube. The calculations were made in the same way as used before. The results are shown in Tables 10 & 11. Two plots were made of 1/U against $\mu^{0.4}/v^{0.8}$ and the extrapolated value of 1/U when $\mu^{0.4}/v^{0.8}$ is zero were found to be (Fig. 17)

At 50° C. of heating water = 0.000362
At 80° C. " " " = 0.000338

Taking the conductivity of copper tube to be 220 B.Th.U./(hr.) (sq.ft.) (°F. Ft.), H_W has a value of 42,000 B.Th.U./hr. Calculated on this basis, the annular film coefficient, H_A, has values

At 50° C. - 2950 B.Th.U./hr.
80° C. - 3200 B.Th.U./hr.

TABLE - 10

Determination of $(1/H_A + 1/H_W)$ for the Copper Tube at 50° C.

Total Heating Area - 1.275 Sq.ft.

Cross sectional area - 0.0012 sq.ft.

Test No.	Feed Rate lbs. /hr.	Feed water Temp. °F. In	Feed water Temp. °F. Out	Heating water Temp. °F. In	Heating water Temp. °F. Out	ΔT °F	Q B.Th.U./hr.	U B.Th.U/hr.	$\frac{1}{U}$	Velocity ft/sec. V	$\mu \times 10^4$ F.P.S	$\frac{DV\rho}{\mu}$	$\frac{V^{0.8}}{\mu^{0.4}}$	$\frac{\mu^{0.4}}{V^{0.8}}$
1.	555	48.0	91.8	122.0	118.0	46.5	24270	409	0.002443	2.062	6.583	7630	33.42	0.029922
2.	608	49.0	91.9	122.0	118.4	45.9	26040	445	0.002246	2.259	6.582	8350	35.96	0.027808
3.	619	48.0	91.0	121.7	117.8	46.0	26600	454	0.002205	2.300	6.615	8460	36.41	0.027466
4.	668	48.2	91.2	122.0	118.2	46.6	28700	483	0.002079	2.485	6.996	8645	37.88	0.026398
5.	722	48.2	90.8	122.0	117.9	46.6	30700	517	0.001934	2.686	6.617	9880	41.21	0.024266
6.	812	47.7	89.6	122.0	117.5	47.4	34000	563	0.001776	3.019	6.695	10900	45.04	0.022198
7.	818	47.8	88.6	122.0	117.0	47.7	33400	550	0.001818	3.047	6.750	10994	45.23	0.022110
8.	853	47.6	88.8	122.0	117.0	47.9	35150	576	0.001736	3.173	6.740	11460	46.74	0.021395
9.	861	48.0	89.1	122.0	117.3	47.6	35380	584	0.001712	3.200	6.700	11620	47.17	0.021200
10.	889	47.3	88.5	122.0	117.2	48.1	36600	598	0.001672	3.305	6.763	11890	48.22	0.020738
11.	905	48.2	88.5	122.0	117.1	47.7	36480	601	0.001664	3.365	6.732	12160	49.01	0.020404
12.	924	47.8	87.9	122.0	116.4	47.8	37060	609	0.001641	3.440	6.740	12440	49.86	0.020055
13.	975	48.0	87.8	122.0	117.8	48.2	38700	624	0.001602	3.625	6.765	13040	53.13	0.018822
14.	1092	48.7	87.3	122.0	116.6	48.0	42140	689	0.001451	4.060	6.750	14640	56.90	0.017575
15.	1094	47.1	86.0	122.0	116.4	49.4	42540	676	0.001479	4.067	6.885	14360	56.53	0.017690

TABLE - 11

Determination of $(1/H_A + 1/H_W)$ at 80° C.

Test No.	Feed Rate lbs. /hr.	Feed water Temp. °F. In	Feed water Temp. °F. Out	Heating water Temp. °F. In	Heating water Temp. °F. Out	ΔT_m °F.	Q B.Th.U./hr.	U B.Th.U./hr.	$\frac{1}{U}$	Velocity Ft./sec. V	μ F.P.S.	$\frac{DV\rho}{\mu}$	$\frac{V^{0.8}}{\mu^{0.4}}$	$\frac{\mu^{0.4}}{V^{0.8}}$
1.	373	46.4	141.5	176.0	171.6	68.20	35460	408	0.002451	1.388	0.0004915	6863	27.37	0.03654
2.	451	55.6	140.2	176.0	171.0	65.80	38110	454	0.002203	1.676	0.0004700	8656	32.41	0.03085
3.	452	48.2	142.6	175.9	170.5	65.66	42700	510	0.001961	1.682	0.0004850	8442	32.02	0.03123
4.	453	47.2	142.4	176.0	170.6	66.52	43100	508	0.001969	1.683	0.0004875	8400	32.05	0.03120
5.	481	46.4	136.4	175.9	170.5	72.20	43290	470	0.002128	1.788	0.0005050	8623	33.16	0.03016
6.	515	47.8	135.3	175.9	170.2	71.70	45050	493	0.002028	1.916	0.0005040	9250	35.07	0.02851
7.	539	48.2	133.8	176.0	169.8	72.60	46140	499	0.002006	2.005	0.0005075	9620	36.27	0.02757
8.	560	46.4	133.5	176.3	170.3	74.40	48780	514	0.001946	2.083	0.0005135	9870	37.22	0.02687
9.	605	47.5	131.0	175.7	169.6	74.80	50500	530	0.001887	2.250	0.0005280	10367	41.95	0.02384
10.	624	48.6	131.5	176.0	169.2	73.58	51720	552	0.001813	2.320	0.0005135	11000	40.57	0.02465
11.	625	46.4	130.7	175.9	169.2	76.00	52700	544	0.001838	2.325	0.0005217	10766	40.39	0.02476
12.	653	47.3	132.5	175.9	169.1	73.30	55630	596	0.001678	2.430	0.0005135	11520	42.10	0.02375
13.	678	46.4	129.8	176.3	169.0	75.86	55940	579	0.001729	2. 497	0.0005250	11560	42.64	0.02345
14.	713	48.9	128.7	176.2	168.7	75.50	56900	592	0.001691	2.653	0.0005208	12400	44.91	0.02226
15.	738	51.7	130.1	176.0	168.4	73.20	57850	620	0.001613	2.750	0.0005080	13164	46.68	0.02142
16.	757	48.2	127.1	176.0	168.0	76.35	59700	613	0.001631	2.815	0.0005280	12980	46.84	0.02135
17.	806	47.2	126.2	176.0	167.9	77.30	63650	646	0.001548	3.000	0.0005240	13935	49.43	0.02023
18.	877	48.2	124.6	175.7	168.3	78.30	67000	671	0.001490	3.260	0.0005358	14820	52.36	0.01910
19.	928	50.8	125.2	176.3	168.2	77.10	69000	702	0.001425	3.452	0.0005255	15920	55.24	0.01811
20.	1005	48.2	123.0	176.1	167.1	78.70	75200	750	0.001333	3.740	0.0005410	16840	58.21	0.01718
21.	1050	46.8	120.5	175.3	165.5	79.60	77400	763	0.001311	3.905	0.0005535	17180	59.60	0.016 78

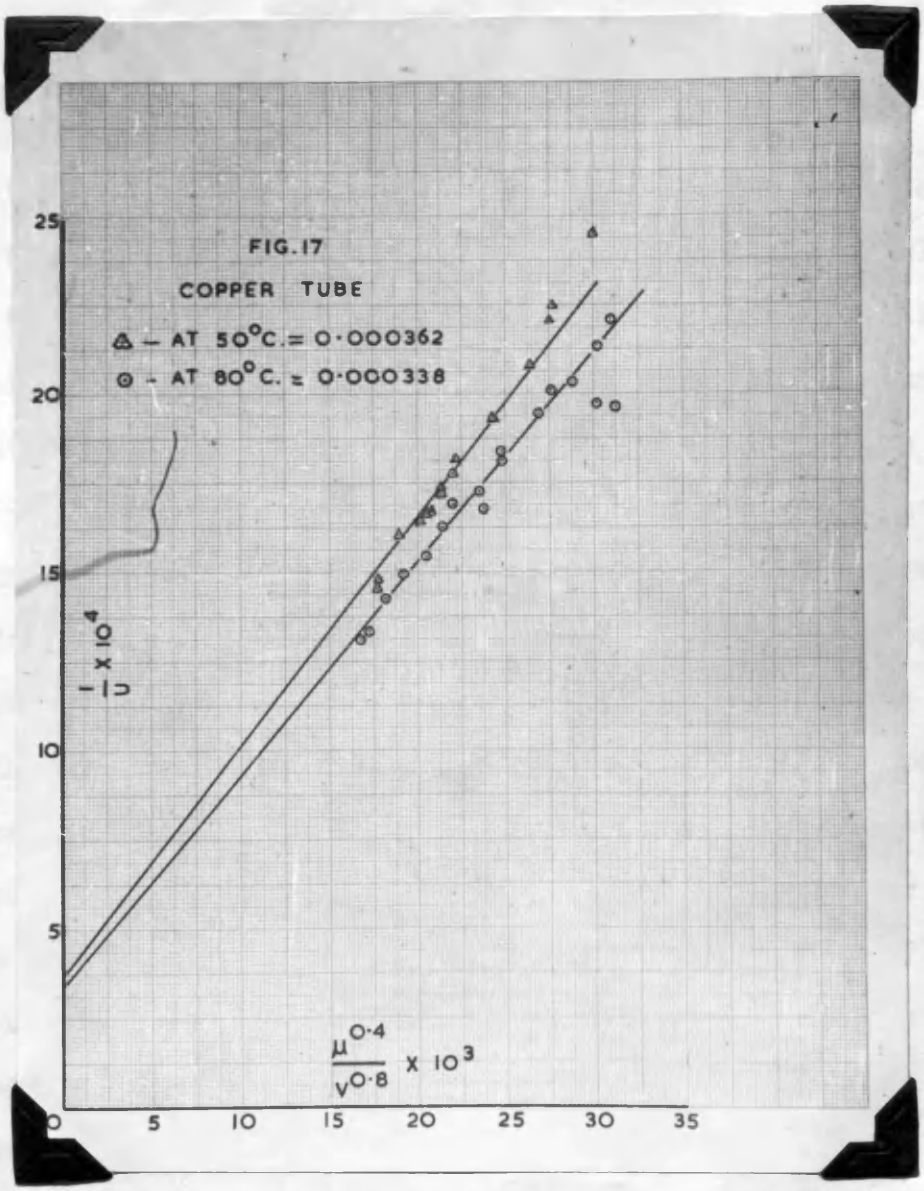


FIG.17
COPPER TUBE
Δ – AT 50°C. = 0·000362
⊙ – AT 80°C. = 0·000338
25
20
15
10
5
0
5
10
15
20
25
30
35
$-\frac{1}{\rho} \times 10^4$
$\frac{\mu^{0.4}}{V^{0.8}} \times 10^3$

FIG. 17

PART I.

SECTION C.

(b) DETERMINATION OF THE BOILING FILM COEFFICIENT OF WATER.

At the top of the copper tube a screw thread was cut so as to fit the brass gland piece used with the stainless-steel tube. This gland piece has a stainless-steel side arm joined by a union nut to the stainless-steel vapour-liquid separator and the condenser system. The thermocouple pocket of 3/32 in. O.D. glass tube was fitted inside the Copper tube through the top of the gland piece. The rest of the apparatus was arranged in the same way as was used in the case of the stainless-steel tube evaporator (Fig. 6).

The first step taken was to standardise the thermocouples. Much care was taken to standardise the thermocouples every time before starting a new set of experiments in order to dispel any error in the determination of difference of temperatures and many times the thermocouples readings were checked in between the experiments of the same set.

For the determination of boiling film coefficient of water, experiments were carried out with cold water at various feed rates ranging from 20 lbs. per hour to 133 lbs. per hour,

a/

a change of about 650% in the feed rates. Some 40 experiments were carried out. The overall heat transfer coefficient, U, was calculated considering the whole length of the tube and the length mean overall temperature drop, ΔT_m, by graphical integration method. U_{NB} and U_B, the overall heat transfer coefficients in the non-boiling and the boiling sections of the tube were calculated. The liquid film coefficients in the boiling section, h_B, and considering the whole length of the tube, h, were also calculated. The results are shown in table - 12.

TABLE - 12

Boiling Film Coefficient of Water in Copper Tube Evaporator.

Test No.	Feed lbs/hr W_f	Evap^n lbs/hr	Feed Temp °F.	T.T.P. Temp. °F.	Max^m Feed Temp. °F.	Q B.Th.U./hr.	ΔT_m °F	U B.Th.U./hr.	Q_{NB} B.Th.U./hr.	ΔT_{NB} °F	U_{NB} B.Th.U./hr.	Q_B B.Th.U./hr.	ΔT_B °F	Boiling Area A_B Sq.ft.	U_B B.Th.U./hr.	h_B B.Th.U./hr.	h B.Th.U./hr.	$\frac{U_B}{U_{NB}}$
1.	20.40	15.94	60.8	160.7	168.8	18020	14.14	1000	2205	59.84	159	15815	10.64	1.0430	1426	3215	1670	8.97
2.	31.73	14.27	54.0	158.5	169.4	17635	16.80	823	3660	35.80	368	13975	11.20	0.9970	1252	2510	1223	3.45
3.	38.49	13.39	66.2	159.9	171.0	17025	13.81	968	4035	27.54	528	12990	10.00	0.9970	1330	2840	1580	2.52
4.	40.58	13.33	52.7	158.3	169.0	17665	17.54	791	4725	40.65	418	12940	11.10	0.9970	1170	2200	1156	2,80
5.	43.46	12.74	51.8	159.1	169.7	17445	17.34	790	5120	41.40	445	12325	10.65	0.9970	1161	2170	1155	2.61
6.	53.38	12.64	54.5	159.1	169.7	18260	16.32	879	6310	37.10	612	11950	10.55	0.9970	1137	2085	1355	1.86
7.	53.58	12.38	54.3	160.2	170.0	18090	18.00	788	6200	46.80	477	11890	9.98	0.9970	1196	2290	1150	2.51
8.	55.09	12.12	52.7	160.2	167.6	18070	19.45	729	6360	51.64	443	11710	10.51	0.9970	1118	2020	1030	2.52
9.	56.43	13.13	44.8	159.5	168.2	19640	19.15	805	6970	47.90	626	12670	11.11	1.0425	1095	1930	1186	1.75
10.	57.28	12.88	45.5	159.1	170.1	19850	18.90	824	7130	54.80	511	12720	9.80	1.0200	1272	2580	1229	2.49
11.	57.72	11.48	46.8	159.5	170.2	18015	17.30	818	7125	42.50	604	10890	10.30	0.9970	1062	1845	1217	1.76
12.	58.27	12.37	48.2	158.4	168.3	18840	20.80	710	7000	43.75	576	11840	11.41	0.9970	1040	1780	999	1.81
13.	61.03	12.13	48.4	160.4	166.5	19020	22.85	658	7205	43.58	457	11815	12.87	0.9270	991	1640	892	2.17
14.	61.10	11.89	54.5	158.9	168.8	18315	19.12	752	6980	42.60	505	11335	10.86	0.9505	1099	1960	1075	2.18
15.	61.84	12.12	48.4	159.5	169.7	19030	16.82	885	7500	41.78	596	11530	9.90	0.9737	1197	2290	1370	2.01
16.	62.54	11.77	56.0	159.1	169.6	18300	19.20	748	7100	51.48	496	11200	10.25	0.9970	1096	1950	1063	2.21
17.	63.45	12.62	51.8	159.1	168.3	19470	18.42	829	7390	41.40	593	12080	10.94	0.9737	1134	2070	1240	1.91
18.	63.88	11.78	51.3	159.2	169.5	18715	18.55	791	7550	41.70	558	11165	10.38	0.9505	1132	2070	1156	2.03

Continued/-

TABLE - 12 (Continued)

Test No.	Feed lbs/hr W_f	Evap^n lbs/hr	Feed Temp. °F.	T.T.P. Temp. °F.	Maxm. Feed Temp. °F	Q B.Th.U/hr.	ΔT_m °F	U B.Th.U./hr.	Q_{N_B} B.Th.U/hr.	ΔT_{NB} °F	U_{NB} B.Th.U./hr.	Q_B B.Th.U/hr.	ΔT_B °F	Boiling Area. Sq,ft. A_B	U_B B.Th.U./hr.	h_B B.Th.U/hr.	h B.Th.U./hr.	$\frac{U_B}{U_{NB}}$
19.	64.88	11.68	56.8	158.9	168.6	18345	20.50	702	7250	46.66	419	11095	10.74	0.9040	1143	2110	1976	2.73
20.	65.76	11.56	46.4	158.5	168.7	18970	20.77	716	8030	48.00	516	10940	10.04	0.9505	1147	2120	1003	2.22
21.	71.02	11.07	51.8	159.2	170.6	18740	19.10	770	8440	38.58	572	10300	10.58	0.8925	1092	1940	1113	1.91
22.	71.69	11.29	51.8	159.1	169.2	19020	19.80	754	8420	41.18	535	10600	10.46	0.8925	1135	2075	1080	2.12
23.	72.17	12.37	46.6	159.5	168.3	20550	18.82	857	8780	43.75	619	11770	10.93	0.9505	1134	2070	1305	1.83
24.	75.32	11.15	45.5	159.5	168.7	19770	24.60	631	9190	47.77	415	10580	11.03	0.8110	1182	2235	844	2.85
25.	77.66	11.76	47.1	158.7	168.6	20470	22.60	710	9430	43.10	572	11040	12.60	0.8925	982	1620	999	1.72
26.	77.71	11.81	47.3	158.4	168.5	20485	20.94	768	9400	45.78	554	11085	11.10	0.9040	1105	1970	1107	2.00
27.	77.80	11.50	46.4	159.7	170.0	20350	19.48	820	9610	41.43	625	10740	9.90	0.9040	1200	2310	1220	1.92
28.	78.10	11.45	50.0	159.9	170.1	20070	19.14	823	9370	43.20	585	10700	9.65	0.9040	1227	2410	1217	2.10
29.	79.93	11.23	49.3	159.5	169.7	20060	20.58	765	9630	42.48	543	10430	9.98	0.8575	1220	2380	1102	2.25
30.	86.66	11.26	47.3	159.8	169.8	21040	20.26	815	10610	42.22	603	10430	9.64	0.8575	1262	2550	1210	2.10
31.	87.60	11.10	45.5	160.2	170.6	21170	23.52	706	10950	47.67	472	10220	9.56	0.7880	1358	2970	984	2.87
32.	87.91	11.21	49.3	160.2	170.4	20995	20.08	820	10650	42.00	608	10345	9.53	0.8575	1267	2570	1220	2.08
33.	88.01	10.96	52.7	160.2	169.8	20450	23.95	670	10300	46.90	452	10150	9.50	0.7880	1358	2970	915	3.00
34.	88.03	11.08	46.4	160.2	170.6	21130	19.88	834	10930	39.83	623	10200	9.23	0.8346	1324	2815	1252	2.12
35.	88.12	10.96	46.4	160.2	169.7	21030	22.30	740	10860	44.30	529	10170	9.31	0.8110	1348	2930	1051	2.55
36.	88.88	10.90	55.4	158.7	169.7	20115	23.44	674	10160	45.92	455	9955	10.24	0.7880	1235	2450	923	2.72
37.	90.08	11.38	46.4	160.2	170.4	21670	21.00	810	11170	40.46	595	10500	9.62	0.8110	1346	2755	1198	2.26
38.	97.62	10.62	35.5	160.0	169.0	21830	22.45	763	12050	39.15	554	9780	9.55	0.7185	1424	3310	1098	2.57
39.	112.08	10.78	46.4	159.7	168.5	23510	26.60	693	13680	42.00	562	9830	10.18	0.6955	1390	3130	959	2.47
40.	133.15	10.05	46.4	160.2	168.8	25240	28.87	686	16290	44.53	526	8950	10.28	0.5790	1505	3790	945	2.86

PART I.

SECTION C.

(c) CORRELATION OF THE RESULTS OF BOILING FILM COEFFICIENTS IN COPPER TUBE AND STEEL TUBE EVAPORATORS.

In the discussion of the paper by Stroebe et al (59) it was commented by Badger - "I hardly think that any set of equation for this (natural circulation) machine will ever be developed that does not contain the term for the feed rate."

Brooks and Badger (13) showed that the feed rate influences the overall heat transfer coefficients in the boiling section of the tube and correlated their results by an equation in which the feed rate was included. Rumford (54) in his paper also reports the influence of feed rate on the boiling film coefficient of heat transfer. The major variable in the present work is the rate of feed, the other variables being dependent. So in the first place a correlation was tried involving the feed rate, but unfortunately in no way a satisfactory means was found to use this term.

In the correlation presented by Stroebe et al (59) the feed rate term is not included indicating that it has no effect on the boiling film heat transfer coefficient. So, the/

the next step taken was to correlate the result of the present work with the Stroebe et al. type equation.

As the equation of Stroebe et al involves data at the average liquid temperature in the tube and as in their experiments the boiling was taking place throughout the entire length of the tube, in the present correlation values were taken at the average temperature of the liquid in the boiling zone. Specific volume of vapour in Cu.ft./lb. was taken from steam table. In evaluating the value of surface tension of water at different temperatures, a linear relationship between the surface tension and the temperature was assumed. The other values of the thermal properties of water at the average liquid temperature in the boiling section were taken from appropriate table.

The heat transfer coefficient of the boiling film was calculated using the equation

$$h_B = \frac{7.8 \times 10^6 \times v^{0.1}}{(C\mu/K)^{0.3} \times \sigma^2 \times (\Delta T_L)^{0.13}}$$

where

h_B = Boiling liquid film heat transfer coefficient, B.Th.U./hr.

v = Specific volume of vapour, Cu.Ft./lb.

σ = Surface tension, dynes/cm.

ΔT_L = Liquid film temperature difference, ^oF.

The/

The values thus obtained were much lower than the actual values. In the present calculations the overall temperature difference, ΔT_B, in the boiling section (the difference in temperature between the heating water and the average temperature of the liquid in the boiling section) was used instead of the true mean liquid film temperature difference. From the data of Stroebe et al (59) it is found that the liquid film temperature differences are approximately 30% of the values of the overall temperature difference. If 30% of the observed overall temperature differences are substituted in the above equation, the values of the boiling film coefficient obtained are still too low. At the same time it was noticed that the values of the observed boiling film coefficient are of the same order as those obtained by Stroebe et al. It will also be noticed from Table - 12 that within the small range of difference in the values of ΔT_B, the boiling film coefficient increases with the decrease in the value of ΔT_B and vice versa as observed by Stroebe et al. So a correlation was attempted on the present results using the entire function of these investigators.

When the function

$$\frac{h_B \left(\frac{CU}{K}\right)^{0.3} \times \sigma^2}{\nu^{0.1} \quad (\Delta T_B)}$$

was/

was plotted against ΔT_B in a log-log paper, a band of points was obtained which could be represented by a straight line as shown in Fig. 18. From this graph the following equation was derived

$$h_B = \frac{8.2 \times 10^8 \times v^{0.1}}{\left(\frac{C\mu}{K}\right)^{0.3} \times \sigma^2 \times (\Delta T_B)^2}$$

When calculated according to this equation, the values of the boiling film coefficients agree quite well with the observed values. Without giving too much weight to the extreme values, which are really doubtful, it will be seen from the Table - 13 that the ratios of the observed and calculated values of the boiling film coefficients are very nearly equal to unity.

The data from the steel tube evaporator were also analysed (Table - 3). When calculated according to the above formula, the boiling film coefficients obtained were too low. It will be noticed from the Table - 3 that under the same conditions of outside heating water and the overhead pressure in both the steel tube and the copper tube evaporators there is a greater overall temperature drop in the boiling section in the former. This is due to the comparatively low conductivity of the steel tube/

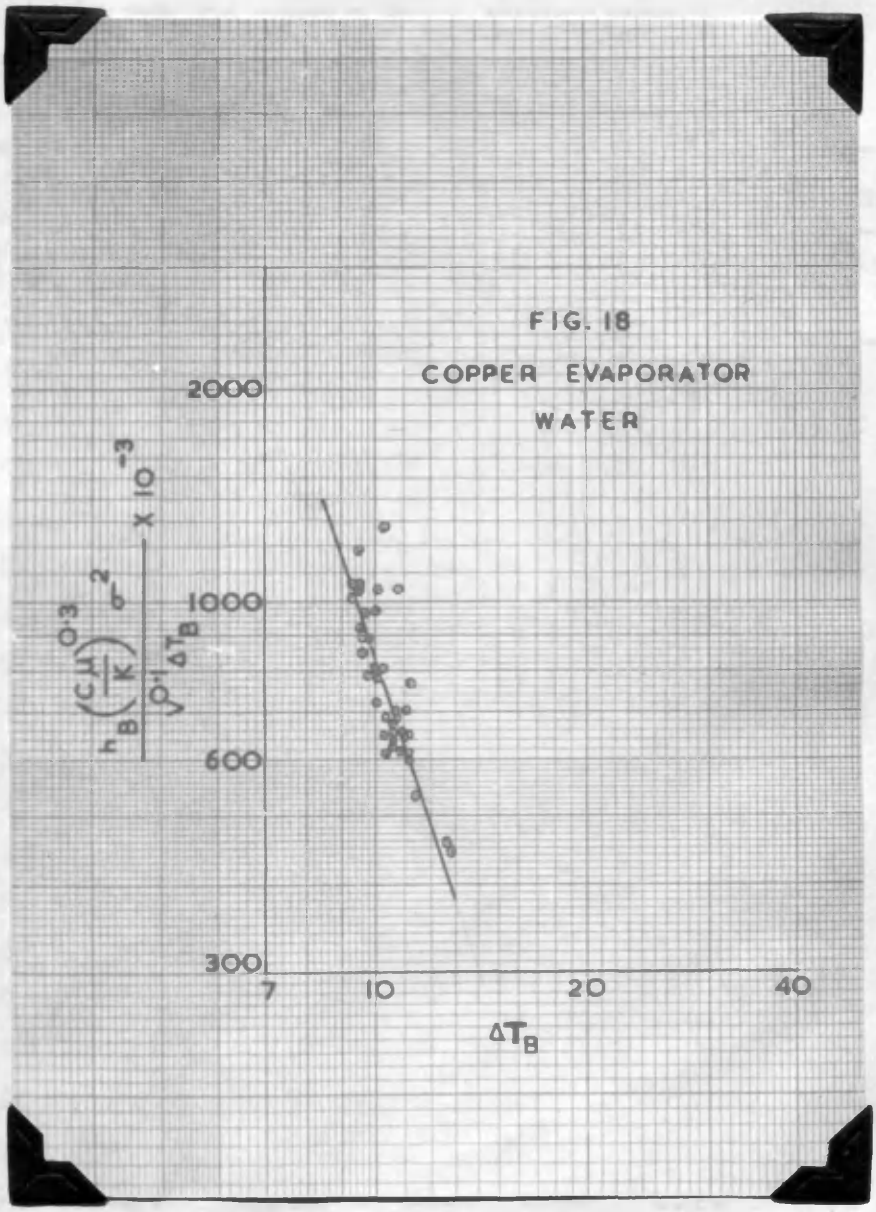


FIG. 18
COPPER EVAPORATOR
WATER
2000
1000
600
300
7
10
20
40
ΔT_B
$h_B \left(\frac{C\mu}{K}\right)^{0.3} \rho^{-2} \nu^{-1} \Delta T_B$
X 10^{-3}

FIG. 18.

T A B L E - 13.

WATER IN COPPER TUBE EVAPORATOR.

Test No.	Sp. vol. of vapour. ft^3/lb v	Surface tension dynes /cm. σ	$\frac{\mu C}{K}$	$\frac{h_B \left(\frac{C\mu}{K}\right)^{0.3} \gamma \sigma^2}{v^{0.1} (\Delta T_B)^3} \times 10^{-3}$	h_B Cal.	h_B Obs.	$\frac{h_B(Obs)}{h_B(Cal)}$
1.	68.61	63.68	2.33	1044	2092	3215	1.51
2.	69.49	63.72	2.35	770	1906	2510	1.32
3.	67.70	63.59	2.32	969	2400	2840	1.18
4.	69.34	63.72	2.35	680	1940	2200	1.13
5.	68.61	63.68	2.33	705	2090	2170	1.04
6.	68.48	63.64	2.32	675	2160	2085	0.97
7.	67.70	63.59	2.31	783	2413	2290	0.95
8.	68.40	63.63	2.32	657	2175	2020	0.93
9.	69.35	63.72	2.35	596	1936	1930	1.00
10.	67.41	63.57	2.31	896	2505	2580	1.03
11.	68.13	63.63	2.33	613	2279	1845	0.81
12.	69.81	63.74	2.37	537	1830	1780	0.97
13.	72.15	63.90	2.45	444	1422	1640	1.15
14.	68.96	63.68	2.345	618	2030	1960	0.97
15.	67.55	63.59	2.34	792	2395	2290	0.96
16.	68.10	63.63	2.33	644	2282	1950	0.86
17.	69.08	63.70	2.36	650	1994	2070	1.04
18.	68.33	63.64	2.34	682	2230	2070	0.93
19.	68.78	63.66	2.34	673	2075	2110	1.02
20.	67.75	63.59	2.32	721	2384	2120	0.89

Continued /-

TABLE - 13 (Continued)

Test No.	Sp. vol of vapour ft^3/lb v	Surface tension. dynes /cm. σ	$\frac{\mu C}{K}$	$\frac{h_B \left(\frac{\mu C}{K}\right)^{0.3} \times \sigma^2}{v^{0.1} (\Delta T_B)} \times 10^{-3}$	h_B Cal.	h_B Obs.	$\frac{h_B(Obs)}{h_B(Cal)}$
21.	68.50	63.64	2.320	628	2140	1940	0.91
22.	68.33	63.65	2.337	680	2190	2075	0.95
23.	69.00	63.65	2.360	650	1998	2070	1.04
24.	69.22	63.66	2.360	701	1950	2255	1.16
25.	70.11	63.86	2.420	457	1484	1620	1.09
26.	69.34	63.72	2.350	609	1940	1970	1.01
27.	67.55	63.59	2.310	798	2440	2310	0.95
28.	67.20	63.57	2.304	850	2585	2410	0.93
29.	67.80	63.59	2.310	812	2410	2380	0.99
30.	67.21	63.57	2.304	900	2585	2550	0.99
31.	67.06	63.55	2.300	1058	2636	2970	1.13
32.	67.03	63.55	2.300	918	2651	2570	0.97
33.	67.00	63.55	2.300	1063	2668	2970	1.11
34.	66.59	63.52	2.290	1037	2830	2815	0.99
35.	66.71	63.53	2.290	1070	2782	2930	1.05
36.	6 8.05	63.63	2.330	808	2282	2450	1.07
37.	67.19	63.57	2.304	975	2597	2755	1.06
38.	67.04	63.56	2.300	1178	2640	3310	1.25
39.	67.95	63.62	2.327	1050	2315	3130	1.35
40.	68.13	63.63	2.330	1260	2270	3790	1.67

tube wall. Thus, though the boiling film coefficients obtained in both the evaporators are almost of the same order, the higher overall temperature drop in the steel tube evaporator tends to lower the film coefficient when calculated using the above equation. It was, however, found that if the exponential value of ΔT_B in the equation was lowered from 2 to 1.75, great harmony could be had between the observed and the calculated boiling film coefficients. It will be noticed from Table - 14 that the ratios of the observed and calculated boiling film coefficients are very nearly equal to unity.

It will be seen that a term for feed rate does not occur in this equation. As suggested by Badger in the discussion of the paper by Stroebe et al (59), perhaps its influence has been masked by the effect of some other variables. In the present correlation the variables - specific volume of vapour, overall difference in temperature in the boiling section, surface tension are all dependent on the independent variable feed rate per hour. In other words the feed rate per hour has influence on the boiling film heat transfer coefficient as surmised by Badger.

It will be seen from Tables - 3 and 12 that under the same conditions of overhead pressure and the temperature of heating water in the jacket, h_B in the steel tube evaporator are lower than/

T A B L E - 14

WATER IN S. STEEL TUBE EVAPORATOR

Test No.	ΔT_B °F	Sp. vol. of vapour. V Ft³/lb	Surface tension dynes/cm. σ	CU K	h_B Cal.	h_B Obs.	$\frac{h_B \text{ (Obs)}}{h_B \text{ (Cal)}}$
1.	16.92	79.10	64.30	2.55	1679	1905	1.13
2.	19.27	83.20	64.49	2.60	1299	1144	0.88
3.	16.92	79.10	64.30	2.55	1679	1812	1.08
4.	18.78	82.30	64.50	2.58	1344	1277	0.95
5.	16.16	77.50	64.20	2.53	1786	1634	0.92
6.	16.54	78.60	64.30	2.54	1708	1559	0.91
7.	15.77	76.85	64.18	2.525	1864	1980	1.06
8.	16.20	77.77	64.22	2.530	1777	1514	0.85
9.	15.55	76.69	64.17	2.520	1912	2188	1.14
10.	16.35	78.29	64.22	2.530	1754	1522	0.87
11.	15.42	76.43	64.15	2.518	1942	1993	1.02
12.	16.36	78.28	64.22	2.530	1754	1644	0.94
13.	14.63	75.33	64.04	2.500	2138	2950	1.38
14.	15.18	76.02	64.12	2.510	2000	2627	1.31
15.	14.91	75.38	64.08	2.500	2065	3817	1.85

than those obtained in the copper tube evaporator. The results of Sauer and Cooper (55) on the better performances of iron for h_B are not substantiated. It should be mentioned here at the same time that the overall difference in temperatures, ΔT_B, were used in these present calculations.

PART I.

SECTION C.

(d) DETERMINATION OF BOILING FILM COEFFICIENTS OF ALCOHOL.

Before starting experiments with ethyl alcohol, the different parts of the apparatus had to be dismantled in order to dry them thoroughly as the alcohol was to be used over and over again.

At this stage it was thought wise to have a check on the value of the combined annular film and wall coefficients, $(1/H_A + 1/H_W)$, both at 50° C. and 80° C. of heating water and to see how these compared with the results obtained just before finding out the boiling film coefficients of water. The results are given in the Tables - 15 and 16. On making the plots of 1/U against $\mu^{0.4}/v^{0.8}$ as usual, the points were found to lie on the same straight line as before, but on carefully drawing the lines the values were found to be

At 50° C. heating water = 0.00045

80° C. " " = 0.00040

It will be seen that there has been an appreciable change in the value of the intercept. The change is, however, in accordance with expectations and can be justified from theoretical/

TABLE - 15

COPPER TUBE EVAPORATOR.

Determination of $(1/H_A + 1/H_W)$ at 50° C.

Test No.	Feed Rate lbs/hr.	Feed water Temp. °F. In	Feed water Temp. °F. Out	Heating water Temp. °F. In	Heating water Temp. °F. Out	ΔT °F.	Q B.Th.U./hr.	U B.Th.U./hr.	$\frac{1}{U}$	Vel. ft/sec. V	μ F.P.S.	$\frac{DV\rho}{\mu}$	$\frac{V^{0.8}}{\mu^{0.4}}$	$\frac{\mu^{0.4}}{V^{0.8}}$
1.	832	48.7	88.3	122.1	118.1	48.37	32930	535	0.001871	3.095	0.0006708	11230	45.91	0.02179
2.	848	48.7	88.2	122.0	118.0	48.35	33480	545	0.001836	3.154	0.0006715	11425	46.59	0.02147
3.	895	47.8	87.3	121.9	117.6	49.00	35350	566	0.001767	3.330	0.0006792	11930	48.43	0.02065
4.	900	48.8	88.1	122.1	117.6	48.17	35370	576	0.001736	3.350	0.0006710	12160	48.90	0.02045
5.	921	47.6	87.4	122.1	117.5	49.00	36650	587	0.001704	3.427	0.0006802	12270	49.53	0.02019
6.	1032	48.7	86.0	122.0	117.0	49.15	38500	615	0.001626	3.835	0.0006812	13700	54.20	0.01845
7.	1053	47.8	85.7	122.0	116.8	49.92	39900	627	0.001595	3.915	0.0006868	13880	54.88	0.01822
8.	1072	48.2	85.9	122.0	117.1	49.50	40400	640	0.001562	3.985	0.0006841	14190	55.74	0.01794
9.	1127	46.8	84.5	122.0	116.9	50.78	42500	657	0.001523	4.192	0.0006975	14630	57.61	0.01735
10.	1151	47.5	84.3	122.0	116.6	50.47	42360	658	0.001520	4.283	0.0006950	15010	58.68	0.01704
11.	1160	47.8	84.7	122.0	116.7	50.13	42800	670	0.001493	4.315	0.0006919	15190	59.14	0.01691
12.	1176	47.5	84.2	122.0	116.6	50.52	43170	670	0.001493	4.374	0.0006951	15310	59.69	0.01675
13.	1263	49.6	84.3	121.9	116.0	49.28	43800	698	0.001433	4.700	0.0006850	16700	63.59	0.01573
14.	1293	48.6	83.7	122.0	116.2	50.20	45370	710	0.001409	4.812	0.0006912	16950	64.56	0.01549
15.	1298	48.7	83.6	122.5	116.1	50.35	45290	706	0.001417	4.830	0.0006912	17000	64.76	0.01545

TABLE - 16

COPPER TUBE EVAPORATOR.

Determination of $(1/H_A + 1/H_W)$ at 80° C.

Test No.	Feed Rate lbs./hr.	Feed water Temp. °F. In	Feed water Temp. °F. Out	Heating water Temp. °F. In	Heating water Temp. °F. Out	ΔT °F.	Q B.Th.U./hr.	U B.Th.U./hr.	$\frac{1}{U}$	Vel. ft/sec. V	μ F.P.S.	$\frac{DV\rho}{\mu}$	$\frac{V^{0.8}}{\mu^{0.4}}$	$\frac{\mu^{0.4}}{V^{0.8}}$
1.	1048	47.8	119.7	176.0	167.0	81.20	75350	728	0.001375	3.896	0.0005527	17180	59.64	0.01677
2.	1054	47.5	120.0	176.1	167.4	81.35	76400	737	0.001357	3.920	0.0005533	17260	59.91	0.01669
3.	1084	47.1	119.2	176.0	167.6	82.20	78100	745	0.001342	4.030	0.0005568	17640	61.10	0.01637
4.	1104	46.6	118.4	176.0	167.0	82.50	79250	754	0.001329	4.105	0.0005610	17820	61.80	0.01618
5.	1134	47.7	118.3	176.0	167.2	82.35	80000	763	0.001311	4.220	0.0005580	18370	63.33	0.01579
6.	1203	47.4	116.1	176.1	166.8	83.80	82500	772	0.001295	4.470	0.0005667	19220	65.90	0.01517
7.	1204	48.2	116.6	176.1	166.6	82.90	82300	779	0.001284	4.474	0.0005616	19400	66.19	0.01511
8.	1224	47.5	115.7	176.2	166.2	83.30	82600	778	0.001285	4.550	0.0005650	19620	66.93	0.01494
9.	1236	47.5	115.1	176.1	166.8	84.45	83500	776	0.001289	4.596	0.0005695	19660	67.25	0.01487
10.	1252	47.7	115.2	176.4	166.4	84.20	84500	787	0.001271	4.656	0.0005690	19910	67.98	0.01471

theoretical standpoint. It can be assumed that there has not been any change in the material of the tube wall which means that the value of $1/H_W$ has remained constant. In that case an increase in the value of $(1/H_A + 1/H_W)$ indicates an increase in the value of $1/H_A$ and consequently a decrease in the value of H_A. In other words, there has been an increase in the resistance to heat flow through the annular film. This is quite possible because in the course of the previous experiments, the outside wall of the tube has picked up some dirt which in turn gives this extra resistance to heat flow. These new values of $(1/H_A + 1/H_W)$ were used in the calculation of the boiling film coefficients, h_B, for alcohol and toluene.

The thermocouples were calibrated with the thermocouples at the inlet and outlet of heating water at 50° C. The apparatus was then fitted as usual for the determination of the boiling film coefficient, h_B, for alcohol.

Altogether 55 experiments were carried out with feed rates ranging from 15 to 85 lbs./hr. which represent a change by more than 500%.

The calculations were done in the same way as described before under glass tube evaporator.

The results are given in the Table - 17.

TABLE - 17

Boiling Film Coefficient of Ethyl Alcohol.

Heating Water at 50° C.

Total Heating Area - 1.275 sq.ft.

Test No.	Feed lbs/hr W_f	Evap^n lbs/hr	Feed Temp. °F.	T.T.P. Temp. °F.	Maxm. Feed Temp. °F.	Q B.Th.U./hr.	ΔT_m °F	U B.Th.U/hr.	Q_B B.Th.U/hr.	ΔT_B °F	Boiling Area. Sq.ft. A_B	U_B B.Th.U/hr.	h_B B.Th.U./hr.	Q_{N_B} B.Th.U/hr.	ΔT_{NB} °F	U_{NB} B.Th.U./hr.
1.	15.66	11.60	50.5	102.9	110.0	4885	18.09	212	4330	15.10	1.1360	252	285	555	42.48	94
2.	15.89	11.32	43.2	103.1	109.6	4840	18.66	204	4217	15.28	1.1128	248	279	623	46.41	83
3.	16.42	10.93	49.1	103.3	110.6	4673	17.62	208	4069	14.72	1.1128	249	280	604	37.17	100
4.	16.89	11.51	55.8	102.7	109.8	4839	17.81	213	4292	15.15	1.1360	250	281	547	39.20	101
5.	18.39	10.07	51.8	103.1	110.5	4407	17.08	203	3761	14.22	1.1360	233	260	646	40.65	114
6.	18.96	10.09	50.5	104.0	110.6	4454	17.41	201	3779	14.22	1.1012	244	274	748	40.18	124
7.	19.19	11.60	43.7	102.9	109.8	5049	18.39	216	4301	15.09	1.1244	254	286	675	37.74	103
8.	19.37	10.09	51.1	103.8	110.7	4452	17.05	203	3762	14.22	1.0896	245	275	690	33.90	110
9.	19.43	10.93	62.3	103.3	110.5	4650	17.03	214	4080	14.90	1.1128	247	277	570	31.50	112
10.	20.24	9.72	55.8	104.4	110.8	4292	16.55	204	3625	13.99	1.1128	233	260	667	33.60	126
11.	21.21	11.16	59.0	102.7	110.1	4812	16.44	230	4160	14.45	1.1128	259	293	652	29.70	136
12.	21.84	10.26	57.2	103.7	110.5	4522	16.98	209	3824	14.59	1.0896	240	270	698	31.24	121
13.	22.32	9.52	53.6	104.2	111.2	4302	16.08	210	3532	13.76	1.0664	241	270	770	27.85	133
14.	24.10	10.90	55.4	102.5	110.1	4834	17.02	223	4043	14.82	1.0425	261	296	791	27.10	126
15.	24.11	9.70	55.0	104.0	111.2	4407	16.28	213	3597	14.08	1.0425	245	276	810	27.00	129
16.	25.19	9.56	56.8	104.0	111.4	4357	15.78	217	3533	13.80	1.0425	246	276	824	25.63	139

Continued /-

Table - 17 (Continued)

Test No.	Feed lbs/hr W_f	Evap^n lbs/hr	Feed Temp. $^oF.$	T.T.P. Temp. $^oF.$	Maxm. Feed Temp. $^oF.$	Q B.Th.U./hr.	ΔT_m ^oF	U B.Th.U./hr.	Q_B B.Th.U./hr.	ΔT_B ^oF	Boiling Area. Sq.ft. A_B	U_B B.Th.U./hr.	h_B B.Th.U./hr.	Q_{NB} B.Th.U./hr.	ΔT_{NB} ^oF	U_{NB} B.Th.U./hr.
17.	26.38	9.87	49.5	104.0	111.7	4591	16.70	216	3611	13.58	1.0200	261	296	980	29.40	131
18.	26.64	9.84	49.1	104.0	111.5	4592	16.28	221	3599	13.48	1.0200	262	297	993	28.20	138
19.	27.10	10.42	50.9	102.9	110.7	4811	17.02	222	3847	14.40	1.0425	256	299	964	28.36	146
20.	28.66	9.20	49.5	104.5	111.7	4418	16.09	216	3355	14.00	0.9970	241	270	1063	24.95	153
21.	28.74	9.83	50.0	103.8	111.7	4639	16.28	224	3579	13.44	0.9970	268	304	1060	26.80	142
22.	29.02	10.40	54.0	103.1	110.9	4815	16.63	227	3829	14.22	1.0425	258	292	986	27.23	156
23.	29.87	10.07	59.0	103.6	111.4	4642	15.66	233	3700	13.61	1.0425	261	298	942	24.94	163
24.	30.04	10.28	50.2	103.6	111.5	4848	16.20	235	3753	13.62	1.0200	270	307	1095	25.80	167
25.	30.09	9.85	55.4	103.6	111.7	4597	16.16	223	3581	13.44	0.9970	268	304	1016	25.93	141
26.	30.44	9.24	59.0	104.7	112.1	4343	15.85	215	3364	13.40	1.0200	246	277	979	24.65	156
27.	31.06	8.19	52.7	105.6	113.0	4080	16.00	200	2957	12.21	1.0200	238	267	1123	31.20	141
28.	31.12	10.22	59.0	103.3	111.9	4709	16.47	224	3716	13.90	0.9970	269	305	993	25.74	139
29.	33.98	9.28	56.3	103.8	112.6	4480	15.92	221	3330	13.08	1.0200	250	281	1150	28.10	161
30.	35.24	8.33	53.1	105.6	113.0	4258	15.42	217	2990	12.00	0.9970	250	282	1268	27.72	165
31.	36.30	8.70	57.2	104.5	112.8	4329	15.79	215	3109	12.90	0.9970	242	272	1220	25.92	170
32.	37.18	9.70	54.9	104.2	112.6	4768	16.20	231	3485	12.84	1.0200	266	302	1283	30.00	168
33.	37.92	8.44	59.0	105.5	113.4	4270	14.15	231	3016	11.70	0.9970	259	293	1254	24.47	184
34.	38.56	9.84	62.6	103.8	112.6	4693	15.93	231	3517	13.00	0.9970	272	309	1176	26.45	160
35.	39.01	8.26	55.4	105.3	112.8	4296	16.16	209	2952	12.13	0.9270	263	298	1344	26.95	144

Continued /-

TABLE - 17 (Continued)

Test No.	Feed lbs/hr. W_f	Evapn. lbs/hr	Feed Temp. °F	T.T.P. Temp. °F	Maxm. Feed Temp. °F.	Q B.Th.U./hr.	ΔT_m °F	U B.Th.U./hr.	Q_B B.Th.U./hr.	ΔT_B °F	Boiling Area Sq.ft. A_B	U_B B.Th.U./hr.	h_B B.Th.U./hr.	Q_{NB} B.Th.U./hr.	ΔT_{NB} °F	U_{NB} B.Th.U./hr.
36.	40.00	8.65	59.0	104.7	112.6	4380	16.90	203	3080	12.44	0.9505	261	295	1300	29.55	136
37.	40.24	8.39	55.4	104.7	113.0	4359	15.81	216	2969	12.40	0.9737	246	277	1390	26.93	175
38.	42.97	8.32	54.5	105.3	113.4	4452	15.73	222	2934	11.88	0.9737	254	286	1518	28.00	180
39.	43.40	9.53	54.7	104.5	113.0	4896	15.81	243	3378	13.28	0.9737	261	296	1518	27.50	183
40.	44.24	9.14	50.2	104.7	112.6	4897	16.01	240	3250	12.18	0.9505	281	322	1647	27.31	186
41.	45.86	8.36	51.1	105.3	113.4	4641	17.20	212	2931	11.72	0.9505	263	299	1710	32.60	162
42.	49.32	9.12	53.2	104.7	113.2	4965	16.71	233	3190	11.87	0.8925	301	349	1796	25.92	158
43.	50.75	8.28	50.4	105.6	113.4	4797	15.97	235	2881	11.36	0.8925	284	326	1916	26.50	189
44.	51.17	8.92	56.3	104.9	112.8	4865	16.16	236	3125	11.97	0.8575	305	353	1740	24.85	168
45.	52.21	8.82	55.4	105.1	112.8	4886	16.71	230	3090	11.88	0.8346	312	363	1775	27.40	170
46.	52.77	8.82	53.6	105.1	112.6	4956	15.81	246	3096	11.82	0.8575	306	354	1860	23.30	191
47.	55.01	7.38	48.2	106.9	114.1	4707	16.86	219	2549	11.17	0.7880	290	334	2158	26.13	170
48.	59.64	8.94	59.0	104.9	112.6	5040	16.36	242	3107	12.20	0.8575	298	344	1933	26.15	177
49.	61.61	7.98	44.2	106.3	113.4	5280	16.90	245	2765	10.87	0.7416	344	406	2515	25.48	185
50.	63.99	7.89	48.7	106.0	113.6	5163	17.25	235	2698	10.94	0.7185	343	406	2465	25.45	174
51.	66.76	7.01	55.4	106.7	114.3	4700	16.67	221	2338	10.66	0.6955	316	368	2362	23.64	175
52.	67.66	7.66	52.2	106.5	113.9	5080	16.63	240	2580	10.51	0.6955	353	420	2500	23.90	181
53.	69.94	7.44	55.4	106.5	113.7	4952	18.00	216	2497	10.73	0.6725	346	410	2455	25.86	158
54.	74.65	7.45	43.3	106.9	113.7	5600	18.15	242	2500	10.73	0.6490	359	428	3100	25.70	193
55.	84.82	8.17	53.6	106.5	113.9	5760	18.44	245	2695	10.72	0.6258	402	491	3065	26.98	175

PART I.

SECTION C.

(e) DETERMINATION OF BOILING FILM COEFFICIENT OF TOLUENE.

The thermocouples were calibrated with the thermocouples at the inlet and outlet of heating water at 80° C. The calibration did not show any appreciable change but the new curve was used to eliminate any possible "drift" errors.

The experiments were carried out with 40 runs in which the feed rates varied from 42 lbs./hr. to about 94 lbs./hr.

The latent heat of vapourisation of toluene at the average temperature at which it left the top-tube-plate was calculated from the Clapeyron-Clausius equation as mentioned in previous case. In the present case it was taken as 175 B.Th.U./lb. The calculation was the same.

The results are given in Table - 18.

TABLE- 18

Boiling Film Coefficient of Toluene.

Heating water at 80° C.

Total Heating Area- 1.275 sq.ft.

Test No.	Feed Rate lbs/hr W_f	Evapn. lbs/hr.	Feed Temp. °F	T.T.P. Temp. °F	Maxm. Feed Temp. °F	Q B.Th.U./hr.	ΔT_m °F	U B.Th.U./hr.	Q_B B.Th.U/hr.	ΔT_B °F	Boiling Area Sq.ft. A_B	U_B B.Th.U/hr.	h_B B.Th.U/hr.	Q_{NB} B.Th.U/hr.	ΔT_{NB} °F	U_{NB} B.Th.U./hr.
1.	42.52	31.60	57.2	156.4	166.1	7329	18.82	306	5339	12.24	0.9970	438	530	1990	42.50	169
2.	43.45	30.92	54.5	157.3	165.8	7321	20.30	283	5244	13.79	0.9970	382	451	2077	43.70	171
3.	44.34	28.28	57.2	156.6	165.1	6837	20.54	261	4780	13.45	0.9505	374	439	2057	40.66	156
4.	44.54	30.60	58.1	156.2	166.3	7220	19.92	285	5150	13.85	0.9970	374	439	2070	43.56	171
5.	47.56	27.83	55,0	157.3	165.8	6952	19.57	279	4687	12.71	0.9270	398	473	2265	37.47	174
6.	48.98	30.10	57.2	156.9	165.8	7355	20.30	284	5070	12.76	0.9970	399	477	2285	41.70	197
7.	49.29	29.65	59.9	156.4	166.5	7220	19.92	285	4965	13.24	0.9737	386	456	2255	38.87	193
8.	50.33	29.53	53.6	156.6	166.7	7383	19.84	292	4935	12.65	0.9737	401	477	2448	42.73	190
9.	51.22	28.72	59.9	156.6	166.1	7140	19.80	283	4803	12.81	0.9505	395	469	2337	39.60	182
10.	52.27	27.11	55.4	157.5	166.1	7025	19.62	281	4540	12.12	0.9505	395	468	2485	40.80	188
11.	52.38	29.24	58.5	157.1	166.1	7320	20.12	286	4998	12.45	0.9737	413	495	2422	44.50	181
12.	52.50	28.78	58.1	157.1	166.7	7256	19.66	290	4805	12.40	0.9505	408	488	2451	40.20	188
13.	54.16	28.24	57.6	157.1	166.3	7245	20.43	278	4717	12.89	0.9040	405	483	2528	37.68	181
14.	54.73	28.80	59.5	156.9	166.3	7320	19.85	289	4810	12.60	0.9040	422	508	2510	37.62	180
15.	54.90	28.05	53.6	156.6	166.5	7325	20.93	275	4660	12.45	0.9270	404	482	2665	43.20	177
16.	55.62	28.10	50.9	157.3	166.7	7443	20.00	292	4673	12.02	0.9270	396	470	2770	42.37	188
17.	56.94	29.36	55.4	158.0	167.0	7635	19.77	303	4905	11.92	0.8925	461	566	2730	37.40	191
18.	57.79	27.55	53.6	157.6	166.7	7390	20.00	290	4510	12.00	0.9040	416	499	2880	39.10	199

continued/

TABLE - 18 (Continued)

Test No.	Feed Rate lbs/hr. W_f	Evapn. lbs/hr.	Feed Temp. ^oF	T.T.P. Temp. ^oF	Maxm. Feed Temp. ^oF	Q B.Th.U./hr.	ΔT_m ^oF	U B.Th.U./hr.	Q_B B.Th.U./hr.	ΔT_B ^oF	Boiling Area. Sq.ft. A_B	U_B B.Th.U./hr.	h_B B.Th.U/hr.	Q_{NB} B.Th.U./hr.	ΔT_{NB} ^oF	U_{NB} B.Th.U./hr.
19.	57.90	28.40	56.1	157.1	166.7	7470	20.23	290	4720	12.24	0.9270	416	499	2750	41.54	190
20.	58.99	27.95	56.3	157.3	166.5	7435	20.43	285	4645	12.38	0.9040	415	498	2790	39.64	190
21.	59.25	27.25	56.3	157.6	166.5	7332	20.30	283	4527	12.22	0.9040	410	491	2805	40.85	185
22.	60.72	26.27	56.7	158.4	166.7	7240	19.76	288	4370	11.82	0.8346	443	539	2870	34.65	188
23.	62.39	29.15	55.0	156.9	166.5	7820	20.24	303	4830	12.66	0.8925	427	516	2990	38.30	204
24.	62.56	27.81	58.1	157.6	166.8	7525	20.54	288	4605	11.75	0.8925	439	533	2920	40.50	189
25.	64.86	25.76	54.5	158.4	167.0	7386	19.40	299	4250	11.22	0.8346	454	555	3136	34.77	205
26.	66.03	28.05	55.4	157.6	167.0	7795	19.37	316	4630	11.96	0.7880	492	612	3165	30.55	213
27.	67.30	26.40	53.6	158.2	167.0	7625	20.65	290	4345	11.61	0.8110	462	566	3280	36.07	196
28.	69.62	25.90	53.6	158.7	167.2	7660	19.92	302	4258	11.25	0.7880	481	595	3402	33.90	206
29.	74.41	26.76	57.6	158.4	167.2	7760	20.35	299	4395	11.25	0.7648	510	641	3365	34.50	191
30.	71.75	28.55	59.5	158.4	167.0	8030	20.07	315	4715	11.47	0.7648	538	685	3315	31.96	203
31.	72.10	27.30	61.3	158.4	167.0	7770	20.40	299	4495	11.92	0.8110	465	571	3275	35.26	200
32.	75.05	27.64	59.0	158.2	167.2	8015	22.13	285	4525	11.49	0.8110	486	603	3490	41.20	183
33.	75.95	27.60	59.4	158.9	167.7	8058	21.05	300	4525	11.32	0.7648	523	662	3 533	35.80	194
34.	76.35	26.82	58.1	158.7	167.4	7977	21.37	293	4390	11.05	0.7880	505	633	3587	38.10	194
35.	80.68	25.08	55.0	159.3	167.7	7990	21.53	291	4080	10.87	0.7416	506	634	3910	36.65	200
36.	83.25	26.62	54.0	159.3	167.7	8405	22.65	291	4335	11.02	0.7185	548	702	4070	37.78	194
37.	84.78	24.38	59.0	159.8	167.7	7915	22.23	280	3955	10.59	0.7185	520	656	3960	37.35	191
38.	90.08	26.48	54.3	159.6	167.5	8690	22.81	299	4695	10.66	0.6490	679	930	3995	37.20	171
39.	91.75	25.85	53.6	160.4	167.9	8710	21.82	314	4200	9.94	0.6955	608	803	4510	36.00	216
40.	93.65	24.15	54.5	160.9	167.9	8490	21.80	306	3925	10.28	0.6490	589	769	4565	31.50	232

PART I.

SECTION C.

(f) CORRELATION OF DATA ON BOILING FILM COEFFICIENTS OF ETHYL ALCOHOL AND TOLUENE.

From the plots of boiling film coefficients h_B against the rates of feed per hour, W_f for both the liquids (Fig.19) it will be found that in the case of toluene the value of h_B increases steadily with the increase of feed rate. In the case of alcohol, the boiling film coefficient is almost unchanged up to a feed rate of about 40 lbs./hr. beyond which rate h_B begins to increase rather abruptly.

From the plots of h_B against the overall temperature drop, ΔT_B, in the boiling section of the evaporator tube (Fig.20), it will be found that there is a steady and gradual decrease of h_B with the increase of ΔT_B in the case of toluene. With alcohol with the increase of ΔT_B there is a gradual decrease of h_B only up to a value of the latter of 300 B.Th.U./hr. When this value is reached, ΔT_B has no influence on the boiling film coefficient.

In an attempt to correlate these results the normal group for liquid properties was introduced and a plot was made of

$$\frac{h_B \left(\frac{C\mu}{K}\right)^{0.3}}{W_f} \quad \text{against } W_f$$

where/

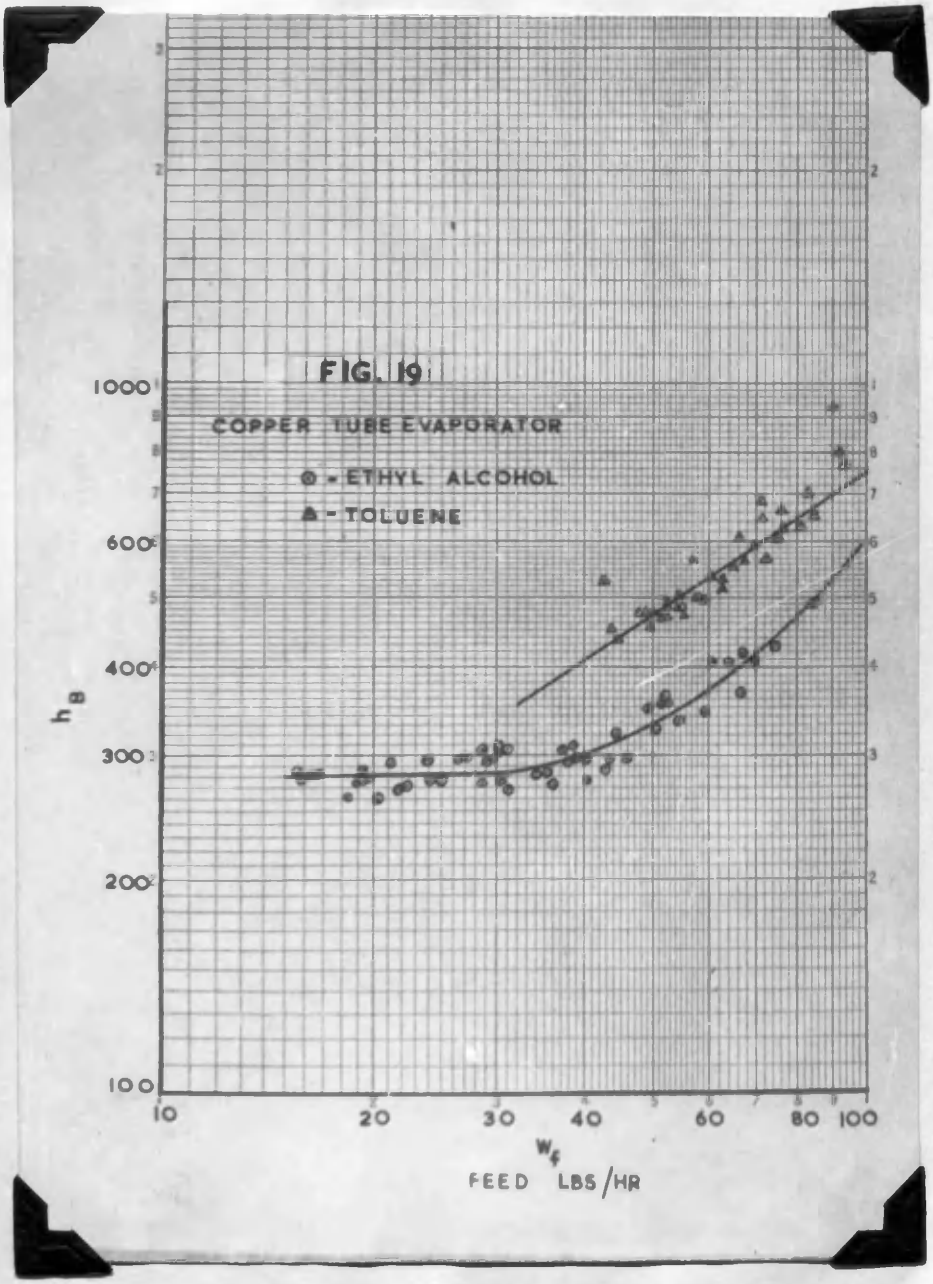


FIG. 19
COPPER TUBE EVAPORATOR
⊙ - ETHYL ALCOHOL
Δ - TOLUENE
1000
600
400
300
200
100
h_B
10
20
30
40
60
80
100
W_f
FEED LBS/HR

FIG. 19

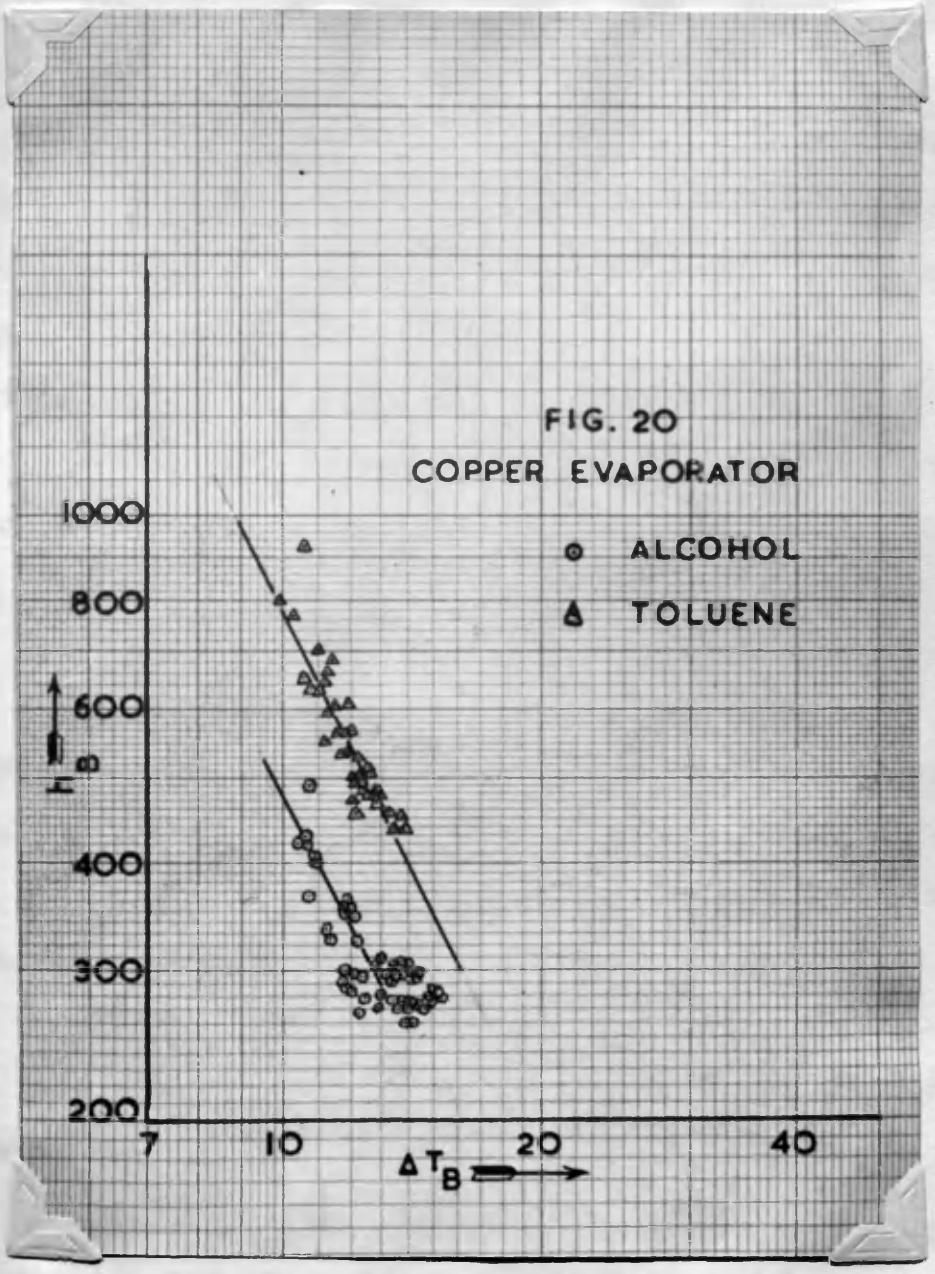


FIG. 20
COPPER EVAPORATOR
ALCOHOL
TOLUENE
1000
800
600
400
300
200
7
10
20
40
ΔT_B

FIG. 20

where

W_f = rate of feed, lbs./hr.

A smooth curve was obtained (Fig.21) which could represent both the results. A straight line can be drawn through the majority of points with a slope of - 0.75. This line mostly passes through the points for alcohol. The line is not shown in the figure.

Next, an attempt was made to find out how much this function is affected by ΔT_B. A graph was drawn to show

$$\frac{h_B \left(\frac{C\mu}{K}\right)^{0.3} \cdot \Delta T_B}{(W_f)^{0.25}} \quad \text{against} \quad \Delta T_B$$

It was found that (Fig.22) the majority of the points for alcohol could be represented, though not very satisfactorily, for lower value of ΔT_B and higher values of W_f, by a straight line having a slope of + 1.85. The majority of the points for toluene, excepting for those with higher values of ΔT_B and lower W_f, were very much scattered and are therefore not shown in the graph.

However, if this straight line is taken as the final correlation for alcohol, an equation of the following type can be worked out,

$$h_B = \frac{30.63\ (W_f)^{0.25}\ (\Delta T_B)^{0.85}}{\left(\frac{C\mu}{K}\right)^{0.3}}$$

When/

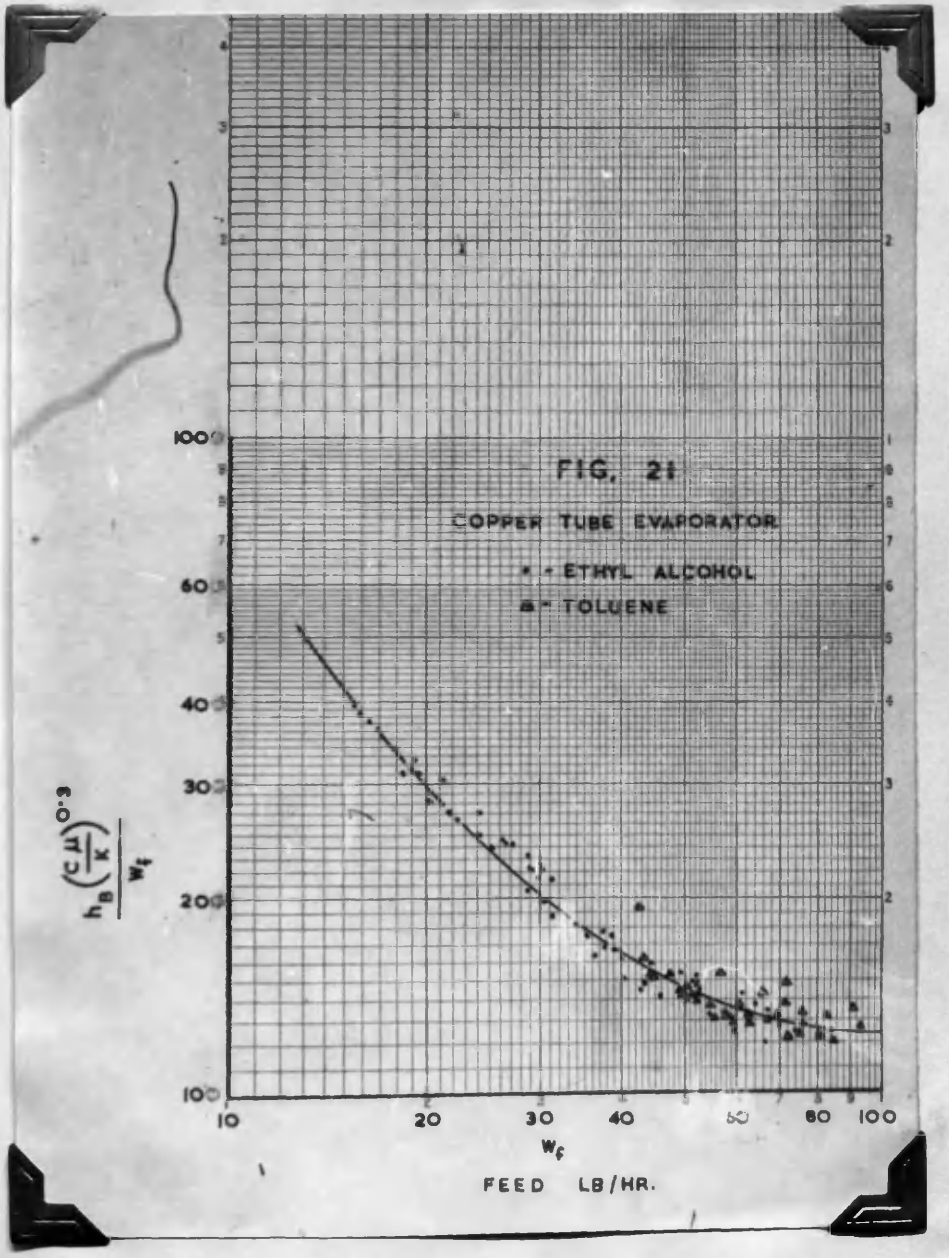


FIG. 21
COPPER TUBE EVAPORATOR
× - ETHYL ALCOHOL
⊞ - TOLUENE
$\frac{h_B\left(\frac{c\mu}{k}\right)^{0.3}}{W_f}$
100
60
40
30
20
10
10
20
30
40
50
60
100
W_f
FEED LB/HR.

FIG. 21.

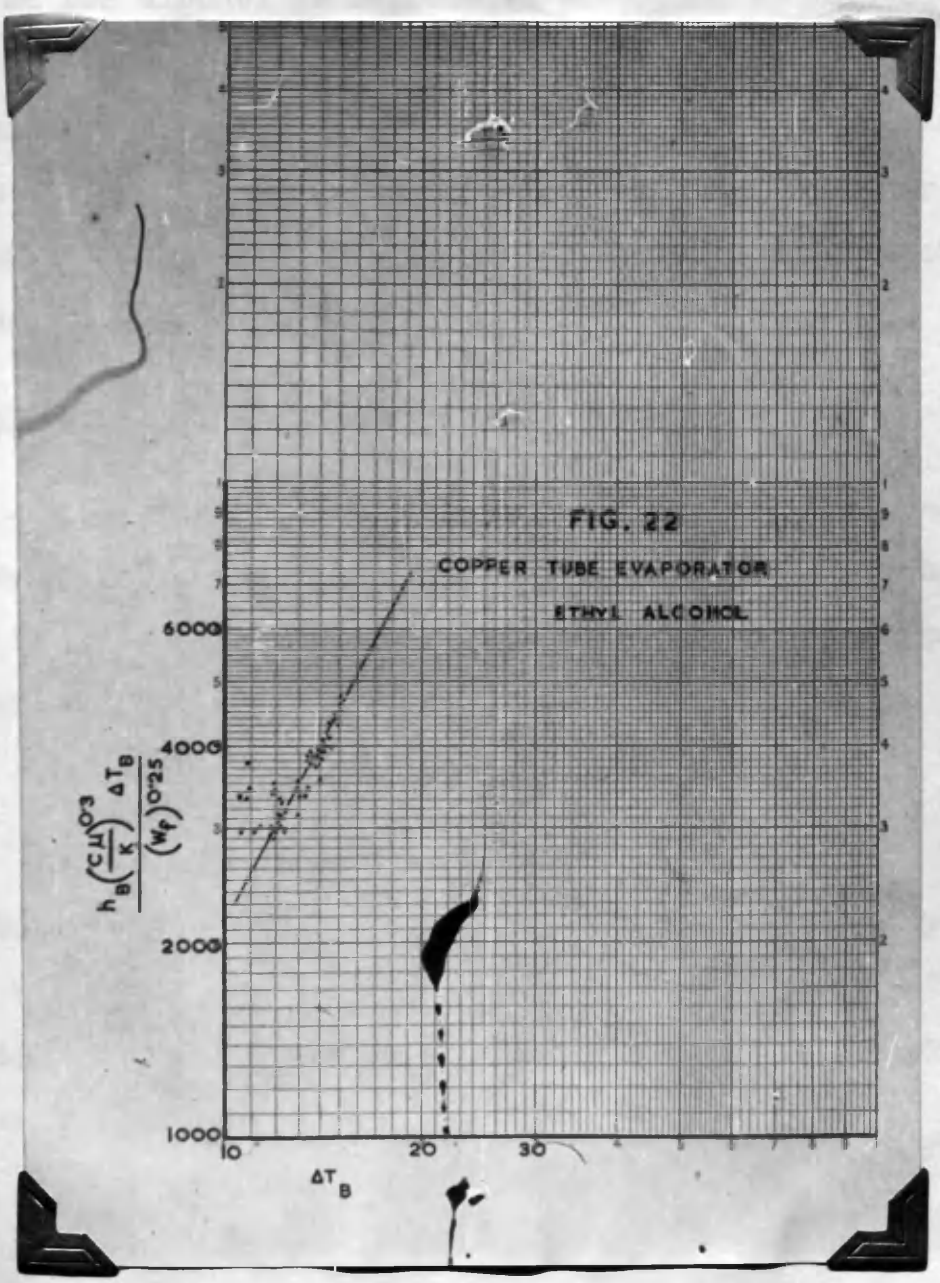


FIG. 22
COPPER TUBE EVAPORATOR
ETHYL ALCOHOL
$\frac{h_B\left(\frac{c\mu}{k}\right)^{0.3}\Delta T_B}{(W_f)^{0.25}}$
6000
4000
2000
1000
10
20
30
ΔT_B

FIG. 22

When h_B for alcohol is calculated according to this equation, the values so obtained tally quite well with the observed values of h_B excepting at very high values of feed rate. The ratio of h_B (observed) to h_B (calculated) is very near to unity and about 76% of the results are within ± 10% of the observed values. The results are shown in Table - 19.

CORRELATION FOR TOLUENE.

A separate correlation of the data for toluene was attempted. From the two plots of h_B against W_f and h_B against ΔT_B (Figs. 19,20) having slopes of + 0.66 and -2 respectively, an equation was developed which has the form

$$h_B = \frac{1696\ (W_f)^{0.33}}{\Delta T_B}$$

The boiling film coefficients so calculated agree very well with the observed values. The results are shown in Table - 20. It will be seen that 92% of the runs fall within ± 10% of the values calculated from this equation.

From an examination of the data in Tables 17 and 18 it will be seen apparently that in both cases the values of h_B increase with increase in W_f and decrease in ΔT_B. The question arises why the equations for h_B will be of different

type/

T A B L E - 19

ALCOHOL IN COPPER TUBE EVAPORATOR.

Test No.	$\frac{\mu C}{K}$	$\frac{h_B\left(\frac{\mu C}{K}\right)^{0.3}}{W_f}$	$\frac{h_B\left(\frac{\mu C}{K}\right)^{0.3}\Delta T_B}{(W_f)^{0.25}}$	h_B Cal.	h_B Obs.	$\frac{h_B(Obs)}{h_B(Cal)}$
1.	13.755	39.88	4737	279	285	1.020
2.	13.764	38.55	4685	282	279	0.988
3.	13.736	37.40	4490	276	280	1.015
4.	13.757	36.50	4620	285	281	0.987
5.	13.711	31.03	3916	276	260	0.945
6.	13.711	31.61	4081	278	274	0.985
7.	13.755	32.73	4528	294	286	0.975
8.	13.711	31.18	4090	279	275	0.987
9.	13.745	31.22	4303	291	277	0.952
10.	13.700	28.16	3756	279	260	0.933
11.	13.723	30.31	4330	290	293	1.012
12.	13.729	27.10	3990	294	270	0.916
13.	13.628	26.52	3752	282	270	0.960
14.	13.741	27.03	4360	305	296	0.971
15.	13.704	25.07	3840	292	276	0.940

Continued /-

TABLE - 19 (Continued)

Test No.	$\frac{\mu C}{K}$	$\frac{h_B\left(\frac{\mu C}{K}\right)^{0.3}}{W_f}$	$\frac{h_B\left(\frac{\mu C}{K}\right)^{0.3}(\Delta T_B)}{(W_f)^{0.25}}$	h_B Cal.	h_B Obs.	$\frac{h_B(Obs)}{h_B(Cal)}$
16.	13.690	24.03	3725	291	276	0.949
17.	13.679	24.54	3880	291	296	1.016
18.	13.674	24.43	3858	290	297	1.025
19.	13.720	24.17	4130	308	299	0.972
20.	13.700	20.65	3580	304	270	0.890
21.	13.672	23.17	3862	295	304	1.032
22.	13.711	22.10	3928	309	292	0.947
23.	13.681	21.82	3790	300	298	0.992
24.	13.681	22.40	3913	294	307	1.046
25.	13.672	22.13	3815	298	304	1.020
26.	13.670	19.90	3453	298	277	0.932
27.	13.611	18.78	3014	276	267	0.967
28.	13.695	21.50	3933	309	305	0.990
29.	13.654	18.10	3330	300	281	0.936
30.	13.600	17.48	2975	282	282	1.000
31.	13.645	16.37	3123	301	272	0.902
32.	13.642	17.78	3436	301	302	1.004
33.	13.585	16.87	3016	281	293	1.044
34.	13.650	17.56	3530	308	309	1.003
35.	13.607	16.68	3156	289	298	1.030
36.	13.622	16.12	3187	300	295	0.985

Continued /-

TABLE - 19 (Continued)

Test No.	$\frac{UC}{K}$	$\frac{h_B\left(\frac{UC}{K}\right)^{0.3}}{W_f}$	$\frac{h_B\left(\frac{UC}{K}\right)^{0.3}(\Delta T_B)}{(W_f)^{0.25}}$	h_B Cal.	h_B Obs.	$\frac{h_B(Obs)}{h_B(Cal)}$
~~37.~~	13.620	15.04	2975	299	277	0.927
38.	13.594	14.55	2900	294	286	0.976
39.	13.664	14.92	3350	323	296	0.917
40.	13.609	15.86	3310	302	322	1.066
41.	13.586	14.23	2940	295	299	1.012
42.	13.594	15.45	3413	305	349	1.140
43.	13.568	14.03	3027	295	327	1.105
44.	13.599	15.09	3375	309	353	1.140
45.	13.594	15.18	3503	308	363	1.179
46.	13.591	14.66	3392	308	354	1.150
47.	13.559	13.25	2990	296	334	1.125
48.	13.610	12.60	3297	326	344	1.054
49.	13.543	14.40	3440	298	406	1.362
50.	13.547	13.84	3422	303	406	1.338
51.	13.533	12.02	2975	300	368	1.226
52.	13.526	13.54	3355	297	420	1.410
53.	13.537	12.80	3318	304	410	1.350
54.	13.537	12.52	3393	309	428	1.386
55.	13.537	12.62	3780	319	491	1.536

T A B L E - 20

TOLUENE IN COPPER TUBE EVAPORATOR.

Test No.	$\frac{h_B\left(\frac{\mu C}{K}\right)^{0.3}}{W_f}$	$\frac{h_B\left(\frac{\mu C}{K}\right)^{0.3}(\Delta T_B)}{(W_f)^{0.25}}$	h_B Cal.	h_B Obs.	$\frac{h_B(Obs)}{h_B(Cal)}$
1.	19.40	3950	478	530	1.110
2.	16.16	3768	427	451	0.947
3.	15.41	3558	441	439	0.996
4.	15.34	3660	429	439	1.023
5.	15.48	3562	477	473	0.992
6.	14.49	3421	480	477	0.994
7.	14.40	3545	464	456	0.984
8.	14.74	3502	489	477	0.976
9.	14.26	3501	485	469	0.968
10.	13.95	3285	516	468	0.907
11.	14.70	3559	503	495	0.984
12.	14.46	3500	505	488	0.966
13.	13.87	3565	491	483	0.984
14.	14.46	3666	504	508	1.008
15.	13.67	3428	511	482	0.943
16.	13.16	3222	531	470	0.885
17.	15.47	3819	540	566	1.048
18.	13.44	3368	539	499	0.926
19.	13.41	3442	528	499	0.945
20.	13.14	3461	525	498	0.948

Continued /-

TABLE - 20 (COntinued)

Test No.	$\frac{h_B\left(\frac{\mu C}{K}\right)^{0.3}}{W_f}$	$\frac{h_B\left(\frac{\mu C}{K}\right)^{0.3}(\Delta T_B)}{(W_f)^{0.25}}$	h_B Cal.	h_B Obs.	$\frac{h_B\ (Obs)}{h_B\ (Cal)}$
21.	12.90	3365	534	491	0.920
22.	13.82	3547	557	539	0.969
23.	12.88	3618	524	516	0.986
24.	13.26	3465	565	533	0.944
25.	13.32	3413	599	555	0.927
26.	14.42	3996	565	612	1.083
27.	13.09	3570	586	566	0.966
28.	13.30	3602	611	595	0.975
29.	13.97	3857	617	641	1.040
30.	14.86	4200	605	685	1.132
31.	12.33	3631	583	575	0.980
32.	12.50	3702	614	603	0.983
33.	13.58	3952	626	662	1.058
34.	12.90	3678	642	633	0.986
35.	12.23	3575	665	634	0.954
36.	13.12	3982	662	702	1.060
37.	12.04	3557	694	656	0.946
38.	16.07	5005	702	930	1.326
39.	13.62	4002	757	803	1.063
40.	12.78	3979	737	769	1.044

type and why in the case of alcohol ΔT_B will have a positive exponent and in the case of toluene it will be negative. On a careful examination of the data, and more evidently from the plots of h_B vs. W_f (Fig. 19) and h_B vs. ΔT_B (Fig. 20) it will be found that in the case of toluene there is a steady increase of h_B with increase of W_f and decrease of ΔT_B and that is why in the equation for toluene ΔT_B has a negative exponent. In the case of alcohol the value of h_B remains almost unchanged up to a certain value of W_f when the value of ΔT_B decreases. Beyond that range h_B increases with increase of W_f but the value of ΔT_B does not decrease appreciably. In the case of alcohol, therefore, both these factors become complimentary and ΔT_B has a positive exponent.

It was realised that the boiling area affects the boiling film coefficient and another attempt was made to introduce this in the final correlation of the data for alcohol and toluene. A plot was made of

$$\frac{h_B \left(\frac{C\mu}{K}\right)^{0.3} A_B}{W_f} \quad \text{against} \quad W_f$$

where A_B = Boiling area, sq.ft.

and a band of points was obtained (Fig.23). A straight line/

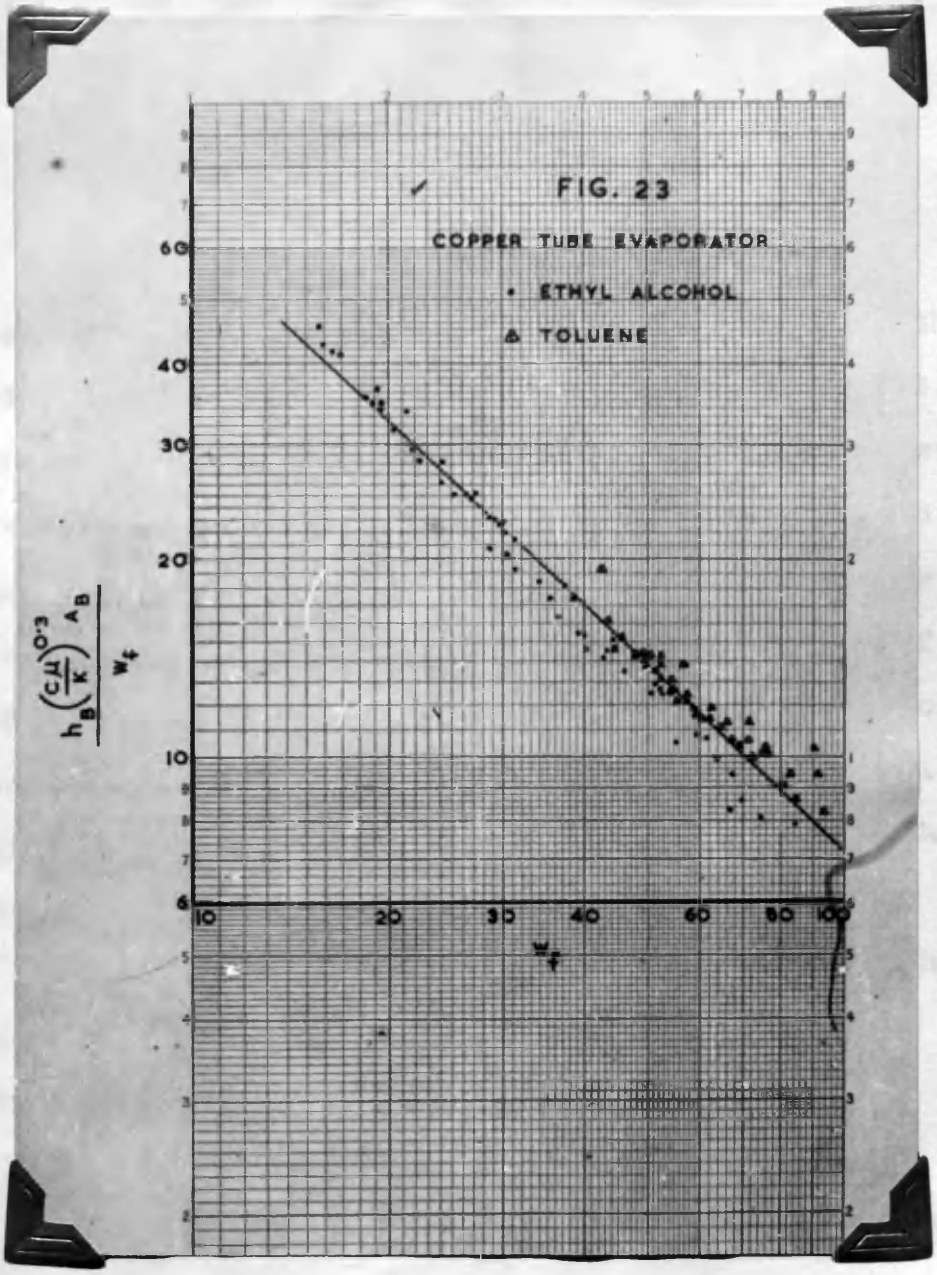


FIG. 23
COPPER TUBE EVAPORATOR
• ETHYL ALCOHOL
△ TOLUENE
$\frac{h_B\left(\frac{c\mu}{k}\right)^{0.3} A_B}{W_f}$
W_f
10
20
30
40
60
80
100

FIG. 23

line drawn through the majority of the points had a slope of - 0.935. An equation

$$h_B = \frac{535\ (W_f)^{0.065}}{\left(\frac{C\mu}{K}\right)^{0.3}\ A_B}$$

was developed from the graph and it holds good for both the liquids, ethyl alcohol and toluene. Boiling film heat transfer coefficients calculated by this equation agree very well with the observed values. 75% of the calculated values of the runs with alcohol are within ± 10% of the observed values and 91% are within ± 15% whereas in the case of toluene 90% of the calculated values are within ± 10% of the observed values of the boiling film coefficients. This equation satisfactorily correlates the data of both ethyl alcohol and toluene. The results obtained are shown in Tables - 21 and 22. This definitely demonstrates the influence of the rate of feed and the boiling area on the boiling film heat transfer coefficient of these two liquids.

T A B L E - 21

ALCOHOL IN COPPER TUBE EVAPORATOR.

Test No.	$h_B\left(\frac{\mu C}{K}\right)^{0.3} \frac{A_B}{W_f}$	h_B Cal.	h_B Obs.	$\frac{h_B \text{ (Obs)}}{h_B \text{ (Cal)}}$
1.	45.29	257	285	1.110
2.	42.86	262	279	1.065
3.	41.60	263	280	1.065
4.	41.44	258	281	1.090
5.	35.22	260	260	1.000
6.	34.80	269	274	1.015
7.	36.80	263	286	1.088
8.	33.95	272	275	1.011
9.	34.72	266	277	1.041
10.	31.33	266	260	0.977
11.	33.70	268	293	1.094
12.	29.50	274	270	0.985
13.	28.25	282	270	0.957
14.	28.16	288	296	1.028
15.	26.12	288	276	0.958

Continued /-

TABLE - 21 (Continued)

Test No.	$\frac{h_B \left(\frac{U\ C}{K}\right)^{0.3} A_B}{W_f}$	h_B Cal.	h_B Obs.	$\frac{h_B \text{ (Obs)}}{h_B \text{ (Cal)}}$
16.	25.05	289	276	0.955
17.	25.02	296	296	1.000
18.	24.90	297	297	1.000
19.	25.17	291	299	1.028
20.	20.60	304	270	0.888
21.	23.08	305	304	0.997
22.	23.00	292	292	1.000
23.	22.72	293	298	1.017
24.	22.84	299	307	1.026
25.	22.05	306	304	0.994
26.	20.28	299	277	0.926
27.	19.15	300	267	0.890
28.	21.43	307	305	0.994
29.	18.45	301	281	0.934
30.	17.42	309	282	0.910
31.	16.31	308	272	0.883
32.	18.12	304	302	0.994
33.	16.82	311	293	0.943
34.	17.50	311	309	0.994
35.	15.47	335	298	0.882
36.	15.32	327	295	0.903

Continued /-

TABLE - 21 (Continued)

Test No.	$\frac{h_B \left(\frac{H\ C}{K}\right)^{0.3} A_B}{W_f}$	h_B Cal.	h_B Obs.	$\frac{h_B \text{ (Obs)}}{h_B \text{ (Cal)}}$
37.	14.64	319	277	0.870
38.	14.17	321	286	0.891
39.	14.52	320	296	0.925
40.	15.07	329	322	0.980
41.	13.52	330	299	0.906
42.	13.79	354	349	0.986
43.	12.52	354	327	0.921
44.	12.93	368	353	0.960
45.	12.66	379	363	0.958
46.	12.57	369	354	0.960
47.	10.43	403	334	0.829
48.	10.80	372	344	0.925
49.	10.69	430	406	0.945
50.	9.94	447	406	0.908
51.	8.36	463	368	0.795
52.	9.41	463	420	0.908
53.	8.60	480	410	0.855
54.	8.12	499	428	0.861
55.	7.90	523	491	0.938

T A B L E - 22

TOLUENE IN COPPER TUBE EVAPORATOR.

Test No.	$\frac{h_B \left(\frac{\mu C}{K}\right)^{0.3} A_B}{W_f}$	h_B Cal.	h_B Obs.	$\frac{h_B \text{ (Obs)}}{h_B \text{ (Cal)}}$
1.	19.33	440	530	1.205
2.	16.10	441	451	1.023
3.	14.64	462	439	0.950
4.	15.29	441	439	0.996
5.	14.34	477	473	0.992
6.	14.44	444	477	1.075
7.	14.02	455	456	1.002
8.	14.34	456	477	1.045
9.	13.56	467	469	1.004
10.	13.25	468	468	1.000
11.	14.30	457	495	1.083
12.	13.74	468	488	1.043
13.	12.54	493	483	0.980
14.	13.08	493	508	1.030
15.	12.66	481	482	1.002
16.	12.21	481	470	0.978
17.	13.80	512	566	1.106
18.	12.15	495	499	1.008
19.	12.43	483	499	1.016
20.	11.88	496	498	1.004

Continued /-

TABLE - 22 (Continued)

Test No.	$\frac{h_B\left(\frac{UC}{K}\right)^{0.3} A_B}{W_f}$	h_B Cal.	h_B Obs.	$\frac{h_B \text{ (Obs)}}{h_B \text{ (Cal)}}$
21.	11.67	496	491	0.990
22.	11.53	538	539	1.002
23.	11.50	504	516	1.024
24.	11.84	504	533	1.058
25.	11.12	540	555	1.027
26.	11.36	573	612	1.069
27.	10.62	557	566	1.017
28.	10.48	575	595	1.035
29.	10.68	594	641	1.080
30.	11.36	594	685	1.154
31.	10.00	560	575	1.020
32.	10.14	562	603	1.075
33.	10.38	596	662	1.110
34.	10.16	579	633	1.093
35.	9.07	617	634	1.027
36.	9.43	638	702	1.100
37.	8.65	639	656	1.027
38.	10.43	726	930	1.282
39.	9.47	679	803	1.183
40.	8.29	728	769	1.056

PART I

SECTION D

KESTNER EVAPORATOR

PART I.

SECTION D.

KESTNER EVAPORATOR.

Work was so far done under reduced pressure and at boiling points of liquids lower than their boiling points at the atmospheric pressure. It was then decided to make some determination at atmospheric pressure. A Kestner climbing and falling film evaporator was chosen for that. Though the Kestner evaporator is the earliest type of those evaporators which are to-day known as "long-tube vertical" evaporators and has been successfully used to secure high rates of heat transfer and to complete the evaporation of thermally sensitive substances in one operation or single pass, there are not many published results available for heat transfer in these units. A considerable number of papers in foreign literature have been written about the merits and faults of the Kestner evaporator but the only work, which has come to the notice of the author, and which makes detailed measurements of heat transfer coefficients, is that of Staub (57).

A working diagram of the evaporator used is given in Fig. (24). In the steam chest there are two copper tubes in/

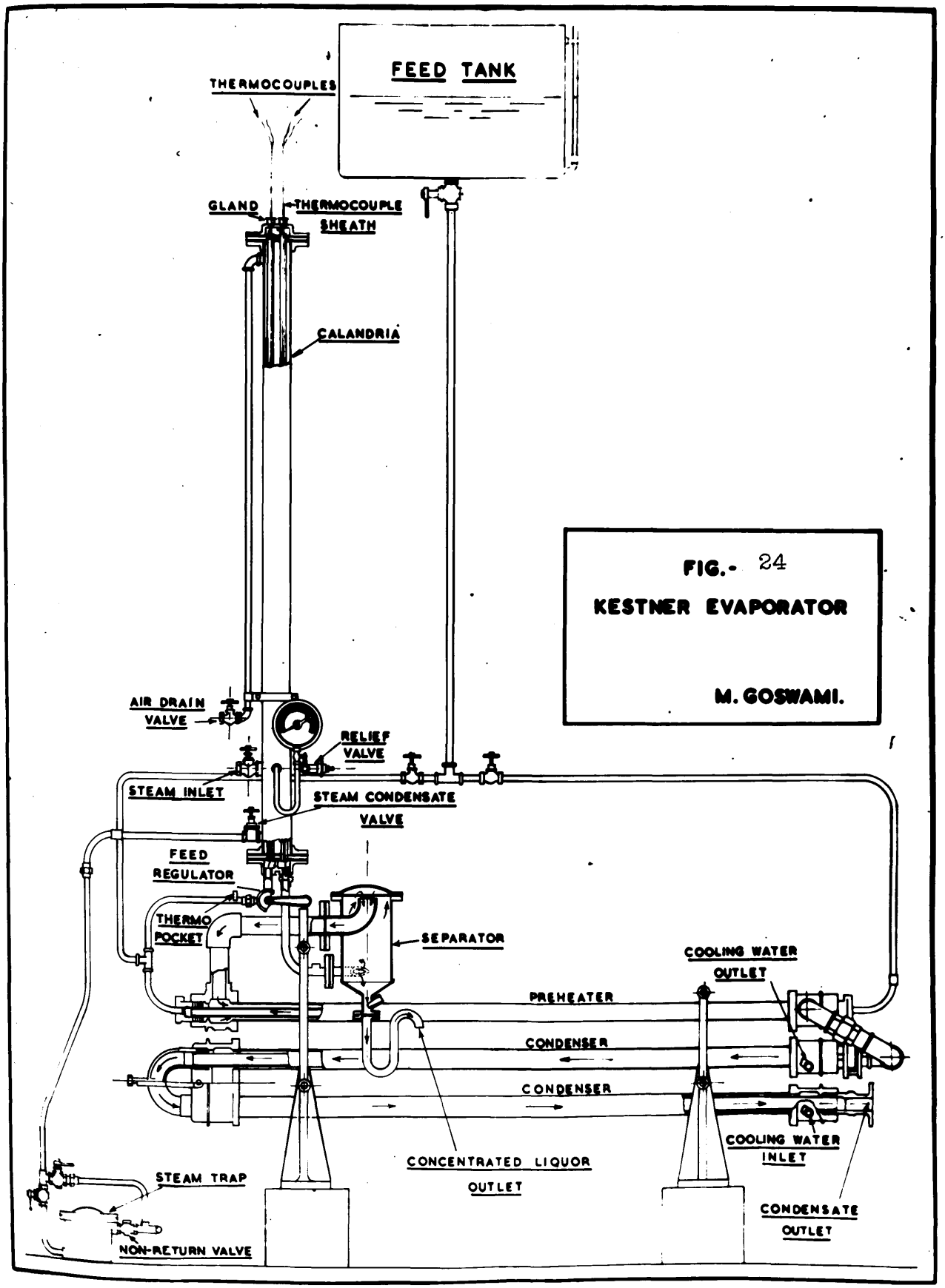


FIG.- 24

KESTNER EVAPORATOR

M. GOSWAMI.

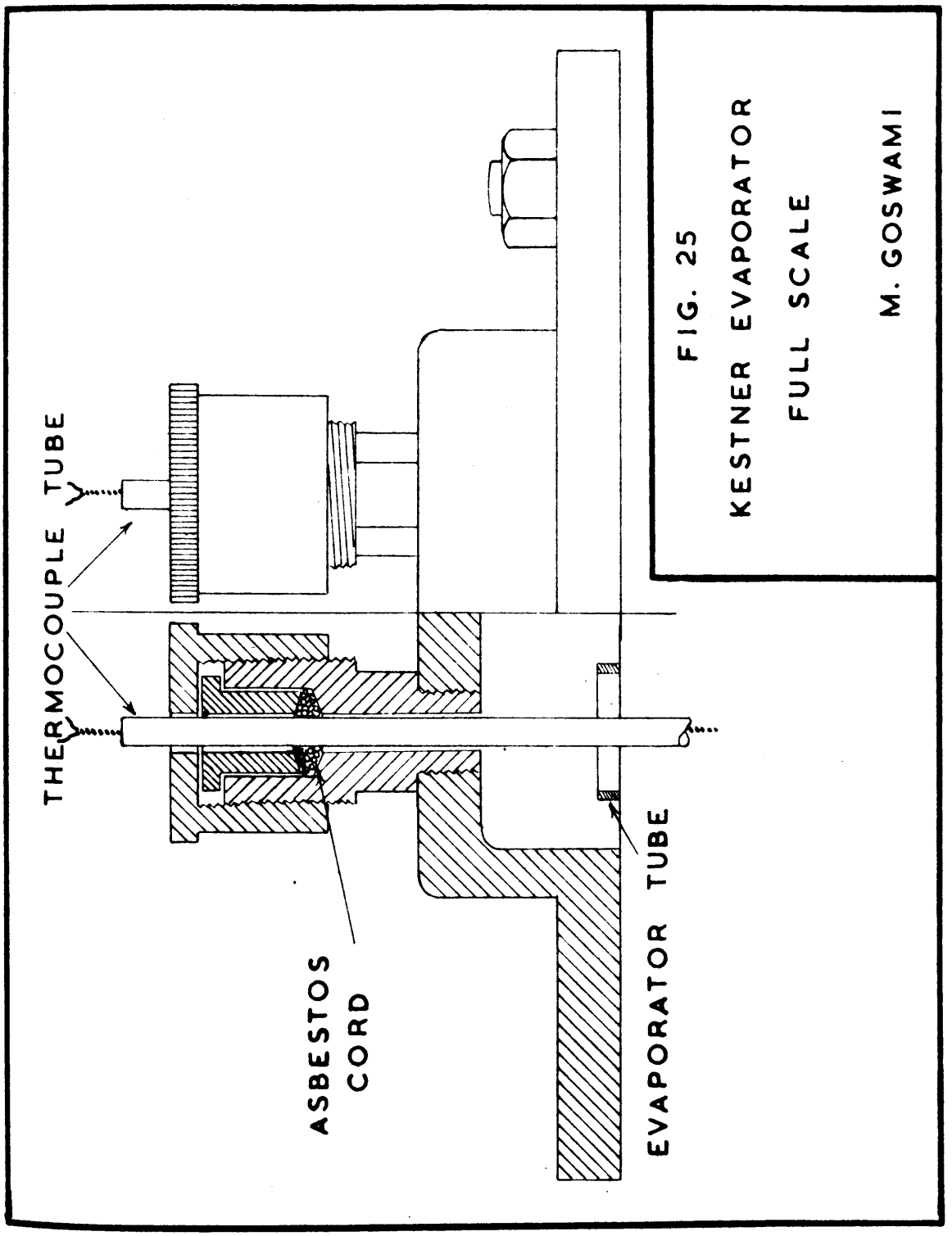
in series, each 6 ft. in length between the tube plates with the external and internal diameters 0.85 and 0.75 inch respectively, giving a heating area of 2.514 sq.ft. There is an overhead feed tank from which the liquid passes through the preheater to the feed regulating valve at the bottom of the first tube through which the liquid flows up. The feed may also be bye-passed, if so desired, without being preheated at all. By carefully manipulating the two valves, the feed may be brought to any desired temperature before being sent through the evaporator tube. The controlled feed passes on into the evaporator proper and flows up the first tube and down the second. Feed liquid is heated to boiling and partly vapourised and the mixture of vapour and liquid enters tangentially into a liquid-vapour separator. Concentrated liquor is discharged through the swan neck tube at the base of the separator, while the vapours pass through the preheater to the condenser and collected as condensate. The steam pressure in the chest is controlled by regulating the steam inlet valve. The steam condensate valve is always full open and is connected to a steam trap. Due regard is paid, as usual, to the removal of non-condensible gases and to be sure, the air drain valve is left just open.

At/

At the top cover of the evaporator, just above the two heating tubes, two gland pieces were fitted so that two thermocouple sheaths of 3/16 inch O.D. of glass tube could be introduced in order to measure the different temperatures of the liquid at different levels inside the evaporator tubes. This is shown in Fig. (25).

The whole body of the evaporator, including the top cover, was well lagged. The body of the liquid-vapour separator was well covered with thick asbestos sheet and then covered with thin foils of pure tin to prevent any loss of heat.

A network of thermocouples was arranged on the same principles as used in the previous experiments. However, in this case there was some difference in that unlike in the previous cases - where one thermocouple junction was kept at the temperature of heating water while the travelling thermocouple was being calibrated - one junction of a thermocouple was kept at a temperature of 0^{o} C. while the travelling thermocouples (two in this case) were being calibrated. In other words, in the previous cases the differences in temperatures between the heating water and the temperatures of the liquid in the evaporator tube at different levels were obtained whereas in the present cases the actual temperatures of the liquid at different levels in the evaporator would be obtained.


THERMOCOUPLE TUBE
ASBESTOS CORD
EVAPORATOR TUBE
FIG. 25
KESTNER EVAPORATOR
FULL SCALE
M. GOSWAMI

PART I.

SECTION D.

(a) DETERMINATION OF COMBINED STEAM FILM AND WALL COEFFICIENTS.

$(1/H_s + 1/H_w)$

For this purpose the liquid-vapour separator was disconnected from the system. The feed exit end of the evaporator was connected through a hose to the drain. The feed tube at the entrance end of the preheater was disconnected from the system and was connected to the main water line. Keeping the pressure of steam at 5 lbs. and 10 lbs./sq. in. in the steam chest, experiments were carried out with various rates of feed recording the temperatures of water at different levels inside the evaporator with the two travelling thermocouples. The temperatures of the feed at the entrance and exit of the evaporator were also recorded with calibrated thermometers. The temperature of water recorded at the exit gave an indication of the change in the rate of feed.

It is worth mentioning here that, unlike the previous cases, the average temperature of the liquid was found out by the graphical integration method of the plot of temperatures vs. distance from the bottom of the evaporator tube. The temperature/

temperature difference was found out by deducting this value of average temperature of the liquid from the temperature of the steam (found out from steam tables).

A typical graph (Fig.26) for finding out the average temperature of the liquid in the evaporator is given (Expt. No. 10 of table - 24). During the plotting of the graph, instead of considering two tubes of 6 ft. in length, a single tube of 12 ft. in length was imagined. The temperatures at the top end of both the tubes should be equal but in fact it was found that the temperature of the liquid at the top of the second tube (through which the liquid flows down) was at a higher temperature than the temperature of the liquid at the top of the first tube. This is explained thus: some heat flows from the steam to the liquid at the top of the evaporator through the partition plate between the steam-chest and the top of the evaporator, suggesting an additional heating area.

The data at steam pressures of 5 lbs./sq.in. and 10 lbs./sq.in. are shown in Tables 23 & 24 and their plots of 1/U against $\mu^{0.4}/v^{0.8}$ are given in Fig. 27. It will be seen that the values obtained at steam pressures mentioned are

5 lbs./sq.in. (227.2° F) = 0.00165

10 lbs./sq.in. (239.4° F) = 0.00150

As/

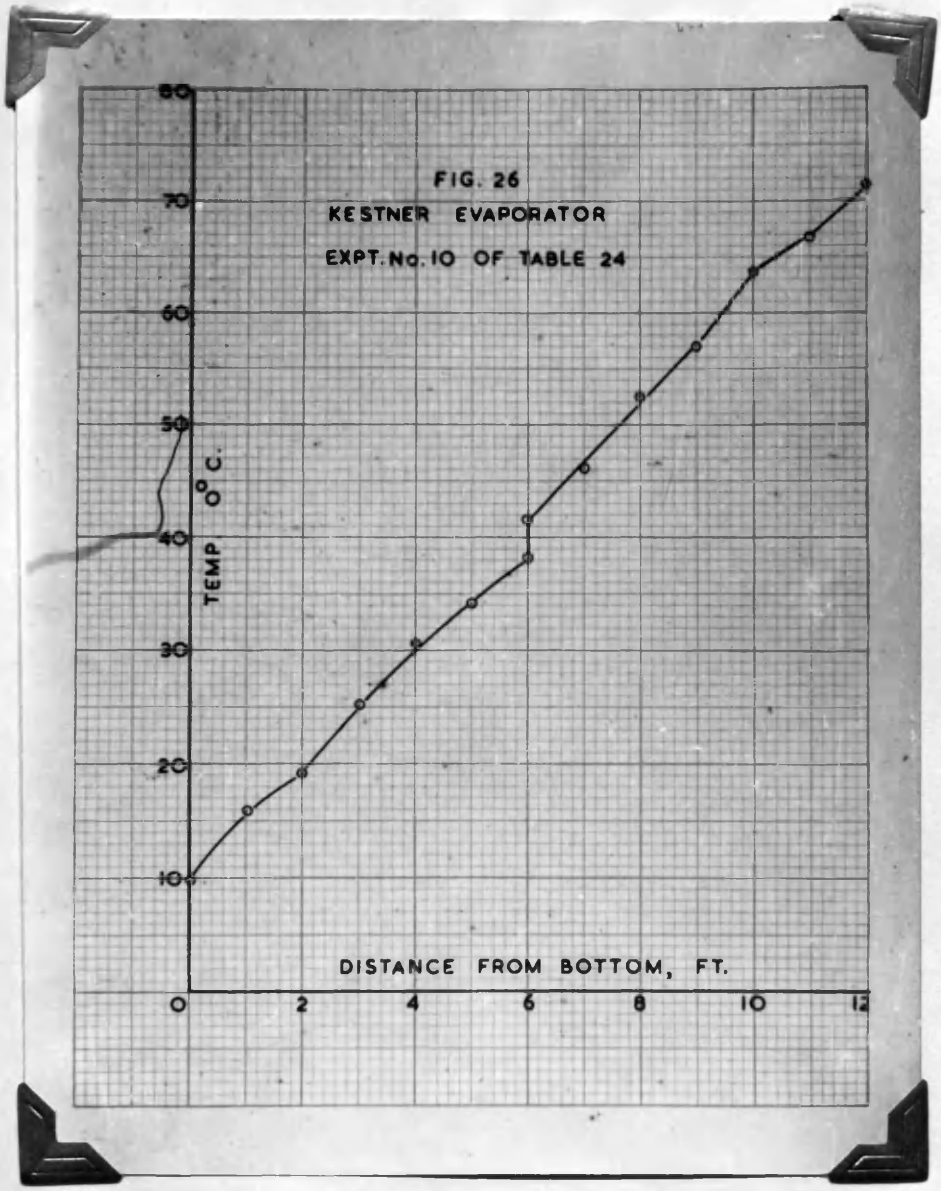


FIG. 26
KESTNER EVAPORATOR
EXPT. No. 10 OF TABLE 24
TEMP °C.
80
70
60
50
40
30
20
10
DISTANCE FROM BOTTOM, FT.
0
2
4
6
8
10
12

FIG. 26

TABLE - 23

KESTNER EVAPORATOR.

Determination of $(1/H_S + 1/H_W)$ at Steam Pr. 5 lbs./sq.in. (227.2° F.)

Total Heating Area - 2.514 sq.ft.

Test No.	Feed water Temp. °F. In	Feed water Temp. °F. Out	ΔT_m °F.	Feed Rate lbs/hr.	Velocity ft/sec. V	Q B.Th.U/hr.	U B.Th.U/hr.	$\frac{1}{U}$	$\mu \times 10^4$ F.P.S.	$\frac{DV\rho}{\mu}$	$\frac{V^{0.8}}{\mu^{0.4}}$	$\frac{\mu^{0.4}}{V^{0.8}}$
1.	48.02	196.9	94.5	522	0.809	77700	327	0.0030581	3.370	9335	20.668	0.048401
2.	52.69	198.7	91.6	528	0.818	77200	336	0.0029890	3.278	9710	21.077	0.047468
3.	48.02	194.6	96.8	582	0.902	85300	351	0.0028490	3.440	10200	22.351	0.044744
4.	48.02	185.2	101.2	666	1.032	91300	359	0.0027831	3.570	11240	25.062	0.039904
5.	51.80	180.4	99.0	723	1.121	92950	374	0.0026774	3.508	13520	26.400	0.037879
6.	51.80	180.7	105.0	739	1.145	95200	361	0.0027718	3.600	12370	26.577	0.037636
7.	48.02	176.6	105.4	768	1.191	98700	373	0.0026825	3.696	12520	27.141	0.036847
8.	52.69	176.3	107.5	825	1.279	102000	378	0.0026455	3.780	13160	28.471	0.035125
9.	51.80	174.2	109.5	852	1.320	104200	379	0.0026385	3.884	13050	28.887	0.034624
10.	52.90	166.1	113.9	979	1.518	112800	394	0.0025381	4.080	14460	31.675	0.031575
11.	50.00	143.8	127.3	1074	1.664	100700	315	0.0031776	4.604	14060	32.480	0.030884
12.	50.36	158.5	121.9	1098	1.702	118800	388	0.0025746	4.357	15210	33.810	0.029577
13.	48.68	147.9	125.8	1330	2.060	131950	417	0.0023958	4.579	17500	38.691	0.025846
14.	50.00	140.0	131.1	1430	2.215	128700	512	0.0019569	4.795	17975	40.170	0.024895
15.	48.92	138.2	131.0	1578	2.445	140600	427	0.0023402	4.790	19840	43.491	0.022993
16.	50.72	141.2	136.0	1594	2.470	144200	422	0.0023686	5.067	18950	42.875	0.023325
17.	48.20	136.9	132.4	1630	2.524	144600	435	0.0023014	4.870	20410	44.320	0.022563
18.	49.82	155.5	122.0	1724	2.670	181900	594	0.0016846	4.364	23790	45.931	0.021772
19.	50.18	132.0	133.5	1779	2.757	145600	434	0.0023095	4.924	21780	46.369	0.021116
20.	40.82	132.0	135.0	1829	2.834	150200	443	0.0022573	5.010	22000	48.090	0.020794
21.	48.55	130.8	135.0	1869	2.896	153800	453	0.0022075	5.005	22510	37.329	0.026796
22.	50.54	131.0	134.5	1894	2.934	152300	449	0.0022272	4.980	22920	49.550	0.020182
23.	48.55	127.3	138.2	2049	3.173	161000	464	0.0021570	5.195	23750	54.563	0.018329
24.	50.00	124.9	139.3	2076	3.215	155500	450	0.0022246	5.265	23750	52.131	0.019183
25.	50.54	128.8	134.6	2090	3.236	163500	484	0.0020674	4.986	25220	53.541	0.018677
26.	50.90	124.6	138.3	2178	3.374	161400	465	0.0021515	5.195	25250	54.488	0.018352
27.	50.36	123.9	137.2	2386	3.695	175700	510	0.0019626	5.136	28010	58.856	0.016992

TABLE - 24

KESTNER EVAPORATOR.

Determination of $(1/H_S + 1/H_W)$

Steam at 10 lbs./sq.in. (239.4° F.)

Total Heating Area - 2.514 sq.ft.

Test No.	Feed water Temp. °F. In	Feed water Temp. °F. Out	ΔT_m °F.	Feed Rate lbs./hr.	Velocity Ft/sec. V	Q B.Th.U./hr.	U B.Th.U./hr.	$\frac{1}{U}$	$\mu \times 10^4$ F.P.S.	$\frac{DV\rho}{\mu}$	$\frac{V^{0.8}}{\mu^{0.4}}$	$\frac{\mu^{0.4}}{V^{0.8}}$
1.	50.00	189.9	112.8	738	1.143	103200	364	0.0027458	3.552	12520	26.681	0.037480
2.	48.42	188.7	111.6	746	1.155	104400	373	0.0026846	3.515	12780	27.015	0.037024
3.	50.90	180.2	120.4	840	1.305	108650	359	0.0027839	3.810	13330	28.833	0.034685
4.	50.00	174.6	119.6	858	1.329	113400	378	0.0026489	3.777	13860	29.370	0.034049
5.	49.82	183.2	117.4	864	1.338	115200	390	0.0025615	3.690	14100	29.806	0.033557
6.	50.00	177.8	120.2	942	1.459	120400	399	0.0025088	3.802	15300	31.564	0.031685
7.	50.00	174.6	114.8	960	1.487	119600	418	0.0023923	3.985	14500	32.166	0.031096
8.	48.74	170.2	128.9	1011	1.567	122700	379	0.0026385	4.150	14660	32.263	0.031000
9.	49.10	164.4	132.9	1236	1.914	142400	427	0.0023446	4.310	17280	36.451	0.027435
10.	47.67	159.8	133.7	1278	1.980	143200	426	0.0023463	4.340	17740	38.212	0.026171
11.	47.30	158.2	132.8	1323	2.048	146600	440	0.0022747	4.303	18530	39.327	0.025432
12.	48.48	153.4	136.7	1395	2.160	146200	426	0.0023474	4.466	18860	40.625	0.024619
13.	47.84	151.0	138.4	1494	2.314	144840	417	0.0023992	4.590	19620	42.335	0.023624
14.	48.20	144.9	141.1	1581	2.448	152840	431	0.0023181	4.678	20410	44.035	0.022712
15.	48.20	143.4	142.8	1608	2.490	152750	425	0.0023507	4.773	20280	44.238	0.022609
16.	48.20	140.3	133.7	1755	2.718	161600	482	0.0020768	4.870	21710	47.022	0.021268
17.	48.20	139.3	144.9	1803	2.795	164200	451	0.0022173	4.886	22240	48.018	0.020829
18.	48.20	139.7	146.3	1815	21812	166000	452	0.0022135	4.963	22050	47.963	0.020851
19.	47.59	137.3	146.0	1864	2.886	167200	456	0.0021944	4.944	22700	49.035	0.020396
20.	46.94	133.9	148.6	1946	3.013	169200	453	0.0022075	5.090	23000	50.071	0.019976
21.	48.38	136.5	146.1	1953	3.023	172050	469	0.0021340	4.950	23750	50.851	0.019665
22.	50.00	130.0	148.2	2194	3.395	175500	471	0.0021222	5.060	26060	55.246	0.018103
23.	46.76	128.8	151.3	2243	3.475	184000	484	0.0020661	5.250	25730	55.540	0.018005
24.	48.20	128.2	151.4	2258	3.497	180600	474	0.0021084	5.254	26350	56.247	0.017781
25.	48.52	124.3	152.7	2423	3.750	182800	477	0.0020986	5.332	27360	58.668	0.017047

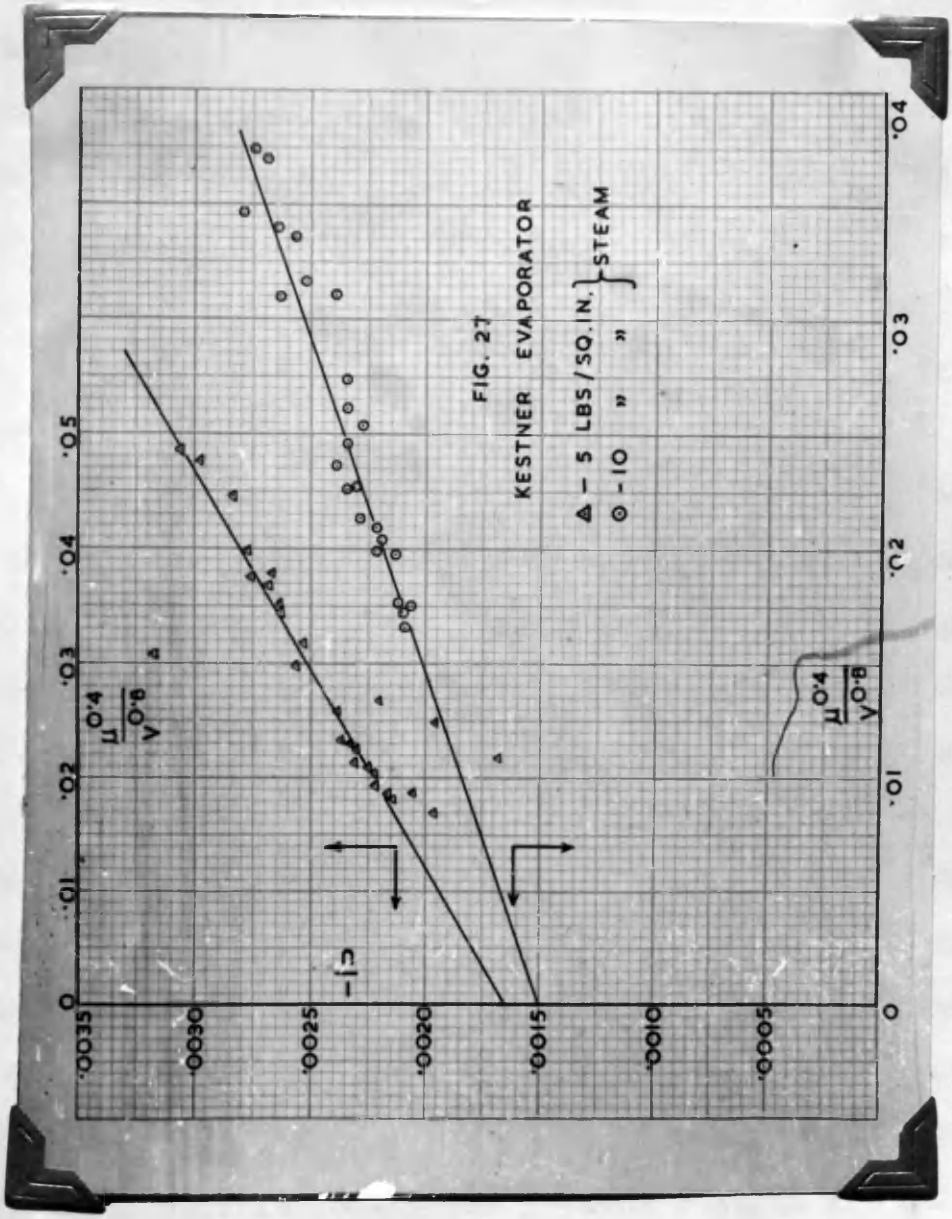


FIG. 27
KESTNER EVAPORATOR
Δ - 5 LBS/SQ.IN. } STEAM
⊙ - 10 " "
$\frac{\mu^{0.4}}{V^{0.8}}$
$\frac{\mu^{0.4}}{V^{0.8}}$
-1:-
0
·01
·02
·03
·04
·05
·0035
·0030
·0025
·0020
·0015
·0010
·0005
0
·01
·02
·03
·04

FIG. 27

As a matter of interest, the results were also calculated using the logarithmic mean temperature difference at steam pressure of 10 lbs./sq.in. The value of the combined steam film and wall coefficients obtained was 0.00155 which is not very different from the result mentioned above. The results thus obtained are not included here. The value of 0.0015 was, however, used for the calculation of the boiling film coefficient in the later experiments.

In the copper tube evaporator, where hot water was used as the heating medium, the values of the combined annular and wall coefficients ($1/H_A + 1/H_W$) obtained at an average difference of temperatures of 75° F. and 47° F. are 0.000338 and 0.000362 respectively, whereas in the present case, where steam has been used as heating medium, at an average temperature difference of 130° F. and 115° F. the values of 0.0015 and 0.00165 have been obtained. As already mentioned check has been made by calculating with the log mean temperature difference in the case of steam at 10 lbs./in^2 pressure. So there cannot be any discrepancy of results from the point of view of calculation. The present results are more or less in keeping with the results obtained in the stainless-steel tube evaporator though in the latter case a low difference of temperatures were maintained. The only possible explanation is/

is that either the tube walls have collected so much of dirt or scale that the resistance to heat flow through them is too large or that the resistance to heat flow offered by the steam film is much greater than that offered by the annular film in the case of hot water circulated at a very high constant velocity through the outer annulus. Calculation of the steam film on the basis of Nusselt's equation suggested an approximate value of 2,000 B.Th.U./hr.

PART I.

SECTION D.

(b) DETERMINATION OF BOILING FILM COEFFICIENT FOR WATER.

The liquid-vapour separator was fitted again and the evaporator was ready for use for the determination of boiling film coefficient.

About 50 experiments were carried out with feed rates varying from 75 lbs./hr. to 407 lbs./hr., representing a change of feed rates by more than 500%. At very high feed rates, the rates of evaporation were very low. So to get a good amount of condensate, the liquid had to be fed at a suitable temperature by manipulating the two valves coming out of the feed tank. The level of water in the feed tank was always kept constant during a run. It used to take about 20 to 25 minutes to reach the steady conditions and a run generally lasted half an hour while the observations were made.

A special difficulty was felt in carrying out the experiments at atmospheric pressure. Due to the changes in barometric pressure, the boiling point of water changed every now and then giving a change in the galvanometer reading. It was therefore necessary to calibrate the thermocouples before/

before each experiment. Much of this laborious work was saved in the following way. By actual experience it was found that there was no change in the thermocouple readings below 95° C. So before the starting of each experiment, the boiling point of water was noted with the thermocouples readings. For each experiment a new plot was made between 95° C. and the boiling point and for any thermocouple reading in between this range this plot was used. For any thermocouple readings below that of 95° C. the original graphs were used.

The data are shown in Table 25.

(c) CORRELATION OF DATA.

In a log-log paper when a plot is made of

$$\frac{h_B \; A_B \; W_E}{W_f \quad T_f} \quad \text{vs.} \quad W_f$$

where h_B = boiling film coefficient, B.Th.U./Hr.

A_B = Boiling area, sq.ft.

W_E = Evaporate, lbs./hr.

W_f = Feed rate, lbs./hr.

T_f = Temperature of feed, °F.

the points lie in a fairly narrow strip.

A/

TABLE - 25

KESTNER EVAPORATOR.

Boiling Film Coefficient of Water at Atmospheric Pressure.

Steam - 10 lbs./Sq.in. (239.4 ^{o} F.)

Total Heating Area - 2.514 sq.ft.

Test No.	Feed Rate lbs/hr. W_f	Evapn. lbs/hr. W_E	Feed Temp. ^{o}F T_f	Aver. Boiling Temp. ^{o}F	Q B.Th.U./hr.	ΔT_m ^{o}F	U B.Th.U./hr.	Q_B B.Th.U/hr.	ΔT_B ^{o}F	Boiling Area Sq.ft. A_B	U_B B.Th.U/hr.	h_B B.Th.U./hr.	Q_{NB} B.Th.U/hr	ΔT_{NB} °F	U_{NB} B.Th.U./hr.	$\frac{U_B}{U_{NB}}$
1.	75.90	26.70	79.10	212.23	35970	35.8	400	25785	27.17	2.0246	469	1586	10185	75.4	277	1.69
2.	103.20	24.00	67.20	211.40	38170	44.1	345	23230	28.00	1.7104	486	1793	14940	79.1	235	2.06
3.	112.50	24.75	66.90	213.13	40420	41.2	391	23880	26.27	1.8500	492	1878	16540	82.1	304	1.62
4.	126.00	25.00	65.10	212.00	42765	43.9	388	24265	27.40	1.7802	498	1999	18500	82.5	306	1.63
5.	128.40	24.00	66.55	212.97	42030	43.9	381	23240	26.43	1.7802	499	1999	18790	82.9	309	1.62
6.	131.40	25.80	123.20	212.65	36750	38.3	382	25010	26.75	1.7450	536	2734	11740	63.5	241	2.35
7.	138.75	22.50	66.55	213.35	42120	45.1	372	21770	26.05	1.6755	499	2000	20290	83.3	291	1.71
8.	139.50	22.50	60.07	211.90	42960	46.9	365	21780	27.50	1.6755	473	1628	21110	85.8	294	1.61
9.	140.55	22.80	99.55	213.92	38030	40.0	379	22020	25.48	1.7802	485	1785	16010	73.8	296	1.64
10.	148.18	23.38	90.10	212.22	40670	41.8	388	22590	27.18	1.7453	477	1677	18080	73.4	321	1.49
11.	117.25	24.75	65.80	212.60	41210	43.6	377	23950	26.80	1.6755	534	2658	17260	77.3	267	2.01
12.	159.90	20.10	66.20	213.44	43000	48.2	355	19380	25.96	1.5359	486	1793	23620	81.9	295	1.65
13.	163.20	18.60	63.20	212.60	42450	49.8	340	18120	26.80	1.4660	461	1496	24330	82.0	278	1.66
14.	176.28	19.08	64.07	212.32	44600	52.6	338	18350	27.08	1.3962	486	1793	26250	83.4	282	1.72

Continued /-

TABLE - 25 (Continued)

Test No.	Feed Rate lbs/hr. W_f	Evapn. lbs/hr. W_E	Feed Temp. ^oF T_f	Aver. Boiling Temp. $^oF.$	Q B.Th.U./hr.	ΔT_m ^oF	U B.Th.U./hr.	Q_B B.Th.U/hr.	ΔT_B $^oF.$	Boiling Area Sq.ft. A_B	U_B B.Th.U/hr.	h_B B.Th.U./hr.	Q_{NB} B.Th.U./hr.	ΔT_{NB} $^oF.$	U_{NB} B.Th.U/hr.	$\frac{U_B}{U_{NB}}$
15.	180.00	17.50	63.20	213.30	43910	51.4	340	17150	26.10	1.2566	522	2418	26760	77.1	277	1.89
16.	189.00	22.50	116.75	212.18	39850	42.5	373	21630	27.22	1.5708	507	2108	18220	66.6	290	1.75
17.	190.50	18.50	64.30	212.30	46150	53.0	347	17910	27.10	1.3614	486	1793	28240	83.3	295	1.65
18.	194.09	17.69	67.35	212.59	45260	53.2	339	16990	26.81	1.2915	491	1868	28270	79.8	290	1.70
19.	200.25	17.25	60.55	212.32	47100	55.5	338	16740	27.08	1.1519	537	2752	30360	80.4	278	1.93
20.	215.85	16.35	65.10	211.82	47470	59.2	319	15920	27.58	1.2217	473	1625	31550	88.9	275	1.72
21.	223.80	15.00	60.56	213.08	48680	61.8	314	14630	26.32	1.0472	531	2610	34040	87.2	267	2.03
22.	227.00	21.00	142.70	212.63	36260	38.5	375	20160	26.77	1.5708	480	1714	16100	57.4	298	1.61
23.	228.75	19.50	118.00	211.82	40320	45.4	354	18800	27.58	1.3612	501	2000	21520	65.8	284	1.77
24.	237.45	21.45	129.70	211.50	40160	42.9	373	20820	27.90	1.4460	510	2162	19340	64.1	288	1.77
25.	258.00	19.80	137.30	213.80	38840	40.3	384	19060	25.60	1.4312	521	2381	19780	60.4	303	1.72
26.	273.75	17.25	114.40	210.80	43030	49.9	343	16530	28.60	1.0471	557	3378	26500	62.4	290	1.92
27.	279.00	16.50	110.50	212.72	44360	52.4	337	16010	26.68	1.0821	555	3313	28350	70.8	280	1.96
28.	282.00	18.00	111.45	211.90	45760	51.4	355	17460	27.50	1.0472	607	6780	28300	68.6	282	2.16
29.	292.35	16.35	107.25	210.90	46280	54.3	340	15980	28.50	1.0821	519	2333	30300	74.1	286	1.81
30.	292.65	16.65	108.30	212.00	44760	54.0	330	14410	27.40	1.0123	520	2363	30350	64.8	312	1.66
31.	293.40	16.20	109.70	212.00	45430	53.4	339	15430	27.40	1.0472	538	2788	30000	72.0	284	1.89
32.	294.70	16.20	115.70	213.60	44360	49.2	359	15890	25.80	1.1519	531	2616	28470	63.7	329	1.62

Continued /-

TABLE - 25 (Continued)

Test No.	Feed Rate lbs/hr. W_f	Evapn. lbs/hr. W_E	Feed Temp. °F T_f	Aver. Boiling Temp. °F.	Q B.Th.U./hr.	ΔT_m °F	U B.Th.U./hr.	Q_B B.Th.U./hr.	ΔT_B °F	Boiling Area Sq.ft. A_B	U_B B.Th.U./hr.	h_B B.Th.U./hr.	Q_{NB} B.Th.U./hr	ΔT_{NB} °F	U_{NB} B.Th.U./hr.	$\frac{U_B}{U_{NB}}$
33.	296.25	14.25	101.30	211.82	46490	58.2	338	13790	27.58	1.0123	494	1910	32700	78.8	277	1.78
34.	297.00	15.75	112.10	212.20	44950	52.9	338	15280	27.20	1.0821	520	2353	29670	72.4	287	1.81
35.	297.15	15.15	107.60	212.00	45660	54.3	335	14580	27.40	1.0123	527	2504	31080	72.1	288	1.83
36.	299.25	16.50	107.60	212.36	47270	53.1	359	16020	27.04	1.0821	548	3078	31250	73.1	299	1.83
37.	316.60	12.60	100.40	212.93	47800	57.2	333	12080	26.47	0.9075	503	2041	35720	73.8	302	1.67
38.	319.95	13.95	95.30	211.60	50740	60.1	336	13540	27.80	1.0123	482	1735	37200	81.9	303	1.59
39.	321.30	13.80	95.30	212.00	50770	59.4	340	13270	27.40	0.9774	496	1938	37500	80.5	304	1.63
40.	322.82	8.52	58.63	213.08	58220	71.0	326	8620	26.40	0.6283	520	2363	49600	86.0	306	1.70
41.	333.30	12.30	91.00	211.40	51940	62.9	329	12040	28.00	0.9076	474	1640	40000	83.0	300	1.58
42.	334.20	12.60	90.30	210.90	52510	64.0	326	12230	28.50	0.5236	819	-	40280	71.1	285	2.87
43.	334.80	10.80	80.20	211.40	54400	67.3	324	10480	28.00	0.3141	1193	-	43920	73.5	272	4.39
44.	349.20	10.80	88.30	212.00	53660	65.2	328	10760	27.40	0.6283	626	10150	42900	77.8	293	2.13
45.	352.95	10.95	79.12	212.00	58500	68.9	338	7715	27.40	0.6283	617	8375	46880	82.8	301	2.03
46.	366.60	9.00	81.28	211.50	56520	69.9	322	8817	27.90	0.5585	567	3770	47700	82.6	296	1.92
47.	398.80	7.80	78.10	211.40	59870	73.5	324	7670	28.00	0.4887	561	3540	52200	83.9	308	1.82
48.	393.64	6.84	74.45	211.10	60336	75.8	317	6636	28.30	0.1047	2242	-	53700	75.8	294	1.65
49.	407.64	6.84	73.53	211.40	62810	77.7	322	6790	28.00	0.3141	773	-	56020	84.4	302	2.56

A straight line which can be made to pass through a maximum number of points has a slope of - 2.5 (Fig.28). An equation is derived from this straight line as follows:

$$h_B = \frac{1.692 \times 10^6 \cdot T_f}{A_B \cdot W_E \cdot (W_f)^{1.5}}$$

The boiling film heat transfer coefficient calculated according to this equation are given in Table (26). It will be evident from this table that the correlation has not been a satisfactory one. At the same time a better correlation of the data was not possible due to the widely variant values of the boiling film coefficients observed.

It is felt that some of the observed values of the boiling film heat transfer coefficient are too high and in a few cases they have reached even unreal values, the reasons for which are not very clearly understood. This has also been noted by Rumford (54). It will be remembered that the boiling film coefficients are calculated from the equation

$$\frac{1}{U_B} - \left(\frac{1}{H_S} + \frac{1}{H_W}\right) = \frac{1}{h_B}$$

The value of $(1/H_S + 1/H_W)$ in the Kestner evaporator with steam at 10 lbs./sq.in. pressure has been found to be 0.0015. So/

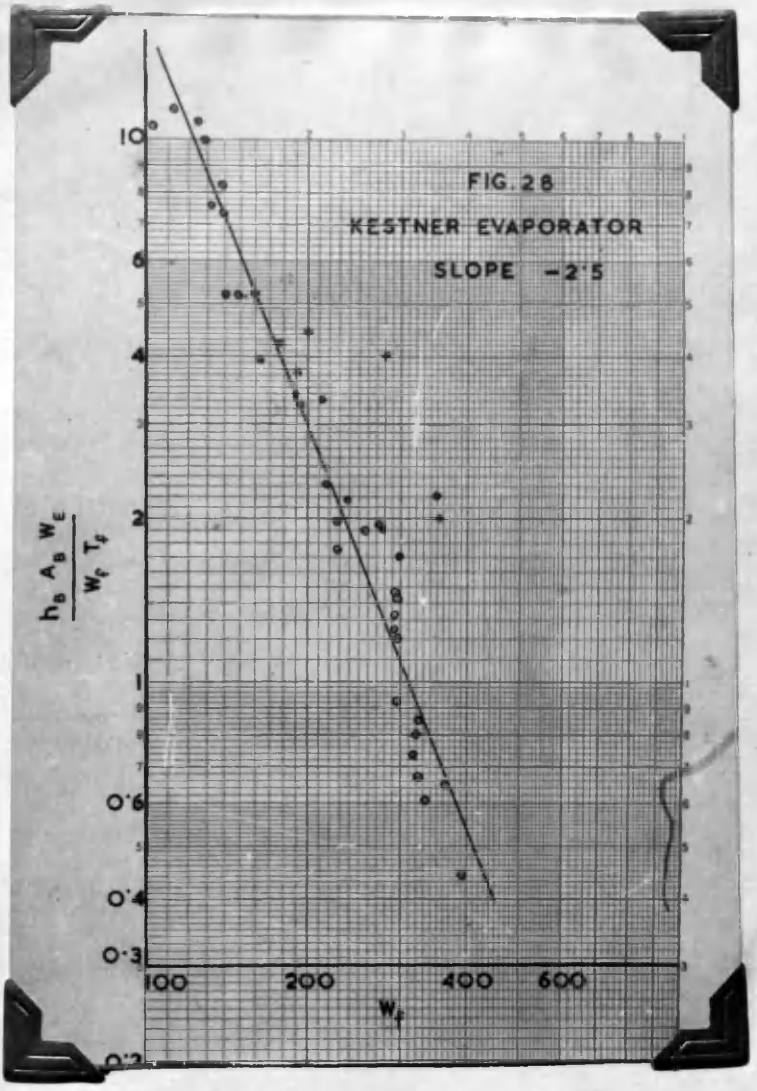


FIG. 28
KESTNER EVAPORATOR
SLOPE -2·5
$\frac{h_B A_B W_E}{W_F T_F}$
10
6
4
2
1
0·6
0·4
0·3
0·2
100
200
400
600
W_F

FIG. 28

T A B L E - 26

KESTNER EVAPORATOR.

Test No.	Vel. of vap. & liq. mix. at exit. ft./sec.	$\frac{h_B A_B W_E}{W_f \cdot T_f}$	h_B Obs.	h_B Cal.	$\frac{h_B(Obs)}{h_B(Cal)}$
1.	69.25	14.260	1586	3741	0.424
2.	62.30	10.620	1793	2644	0.677
3.	64.20	11.430	1878	2072	0.907
4.	64.90	10.850	1999	1753	1.141
5.	62.30	10.000	1999	1771	1.130
6.	67.00	7.600	2734	3074	0.890
7.	58.42	8.230	2000	1829	1.094
8.	58.45	7.320	1628	1639	0.993
9.	59.20	5.200	1785	2495	0.715
10.	60.80	5.205	1677	2071	0.810
11.	64.25	14.300	2658	2116	1.257
12.	52.25	5.225	1793	1795	0.999
13.	48.50	3.995	1496	1884	0.795
14.	49.65	4.235	1793	1740	1.032
15.	45.55	4.670	2418	2016	1.200
16.	58.50	3.380	2108	2152	0.981
17.	48.20	3.750	1793	1646	1.089
18.	46.10	3.264	1868	2276	0.821
19.	44.95	4.470	2752	2292	1.202
20.	42.65	2.305	1625	1743	0.934
21.	39.20	3.030	2610	1950	1.340
22.	54.75	1.746	1714	2139	0.802
23.	50.80	1.966	2000	2175	0.920

Continued /-

TABLE - 26 (Continued)

Test No.	Vel. of vap. & liq. mix. at exit. ft./sec.	$\frac{h_B A_B W_E}{W_f T_f}$	h_B Obs.	h_B Cal.	$\frac{h_B(Obs)}{h_B(Cal)}$
24.	55.90	2.178	2162	1890	1.144
25.	51.60	1.902	2381	1980	1.203
26.	45.00	1.950	3378	2368	1.426
27.	41.85	1.936	3313	2322	1.428
28.	47.00	4.060	6780	2171	3.140
29.	42.80	1.318	2333	2054	1.136
30.	43.60	1.258	2363	2169	1.090
31.	42.40	1.468	2788	2180	1.279
32.	42.40	1.430	2616	2075	1.260
33.	37.40	0.920	1910	2133	0.895
34.	41.20	1.203	2353	2173	1.084
35.	39.70	1.199	2504	2319	1.080
36.	43.20	1.706	3078	1972	1.562
37.	33.10	0.734	2041	2637	0.775
38.	36.55	0.803	1735	1698	1.022
39.	36.20	0.669	1938	2078	0.932
40.	22.60	0.853	2363	3198	0.740
41.	32.35	0.605	1640	2271	0.723
42.	33.15	-	-	-	-
43.	28.45	-	-	-	-
44.	28.48	2.230	10150	3379	3.050
45.	28.90	2.060	8375	2940	2.850
46.	23.90	0.652	3770	3902	0.967
47.	20.80	0.440	3540	4461	0.795
48.	18.34	-	-	-	-
49.	18.36	-	-	-	-

So if the value of $1/U_B$ is less than 0.0015 or the value of U_B is more than 667 B.Th.U./Hr. then h_B reaches an unreal negative value. A very small amount of error in the calculation of the value of U_B may amount to a very great error in the value of h_B. From an examination of the boiling film coefficients of water in steel-tube, copper-tube (Tables 3,12) and Kestner (Table 25) evaporators it will be seen that the values are much more varied in the Kestner evaporator than in the other two. This discrepancy is more pronounced at higher rates of feed where the boiling lengths concerned are comparatively small and a small error in the determination of the boiling section leads to a large error in the value of the boiling film coefficient. Another possible reason for this erratic values of the boiling film coefficients may be due to the uncertain nature of the type of condensation of the steam because the calculations are based on the assumption that the annular film coefficient and the wall coefficients are constant. This effect has been altogether dispensed with by an almost constant value of the annular film coefficient, and has given no troubles in the previous sets of experiments.

Plots of overall heat transfer coefficients against pounds of evaporate per hour (U_B vs. W_E) for water in steel tube/

tube, copper tube and the Kestner evaporators (Fig.29) show that while U_B increases considerably with the decrease of W_E in the steel tube and copper tube evaporators, it is not very perceptible in the case of Kestner evaporator. Brooks and Badger (13) observed a notable change of the overall boiling film coefficient with evaporation in lbs./hr. under reduced pressure. An increase in the amount of vapour produced possibly keeps a greater portion of the boiling section dry and thereby decreases the overall heat transfer coefficient, On the other hand it also increases the velocity of flow in the boiling section counter-affecting the previous effect. The conditions of experiments in the steel tube and copper tube evaporators were the same as in the glass tube evaporator in which boiling in spurts was observed. This possibly explains the chance of keeping the tube dry and the net result obtained of the two above factors is to lower the value of U_B at higher values of W_E. In the case of the Kestner evaporator used this effect is not very much pronounced possibly because of the fact that there is always a falling film of liquid in the second section which does not give the boiling section a chance to become dry. The work of Staub (57) in Kestner evaporator also shows that the various velocities of vapour at the exit of the tube does not very much alter the overall boiling film coefficient.

The/

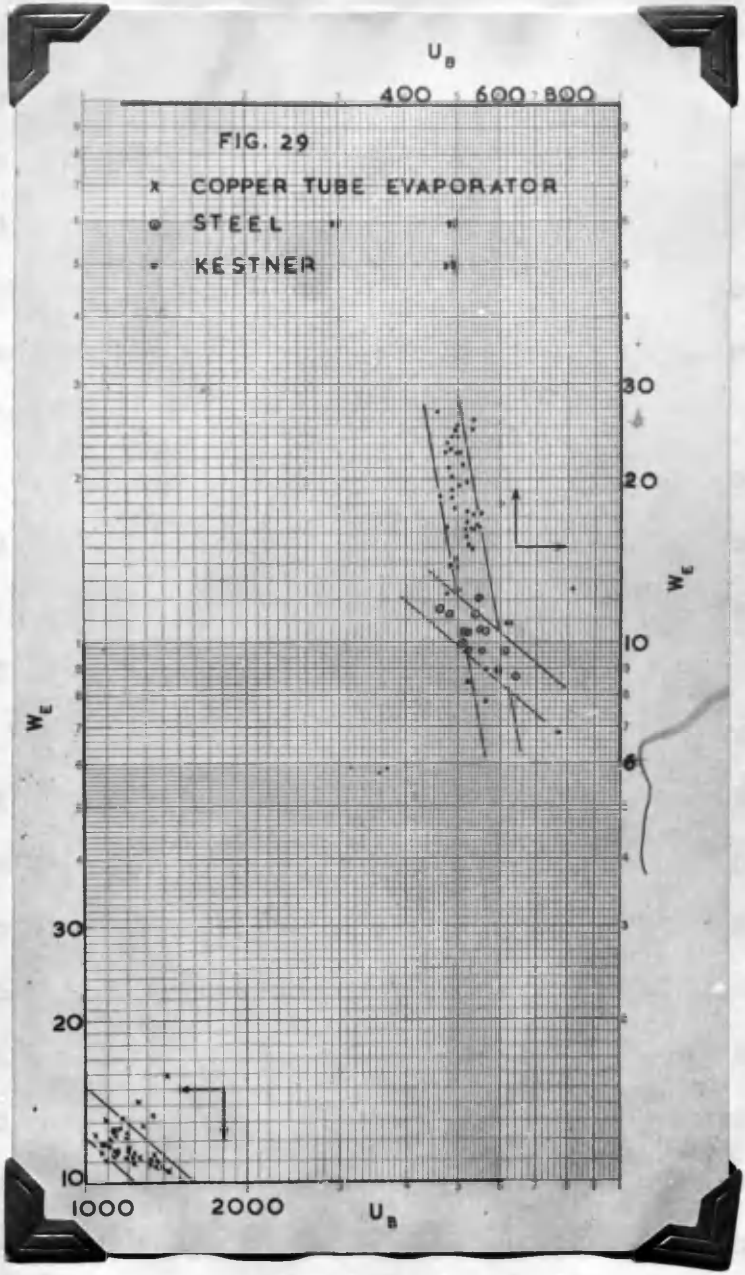


FIG. 29
COPPER TUBE EVAPORATOR
STEEL
KESTNER
U_B
400
600
800
1000
2000
U_B
W_E
30
20
10
W_E
30
20
10
6

FIG. 29

The overall heat transfer coefficient in the boiling section of the steel tube and the Kestner evaporators are round about 500 B.Th.U./hr. while in the copper tube evaporator on an average it is about 1200 B.Th.U./hr. and goes up to 1500 B.Th.U./Hr. Staub (57) gives a value of U_B in the Kestner evaporator of about 1200 B.Th.U./Hr. which in some cases reach as high as 1440 B.Th.U./Hr. In the present work it has already been mentioned that the combined values of the annular film and the wall coefficients obtained are lower and of the order obtained in the stainless steel tube evaporator. So, in spite of the copper tube in the Kestner evaporator, the values of the overall heat transfer coefficient in the boiling section, U_B, obtained are lower and are of the same order of those obtained in the steel tube evaporator. This is perhaps due to some sort of deposit outside the tubes of the Kestner evaporator. In the copper tube evaporator, however, the overall boiling film coefficients, U_B, are much higher and are of the same order of those obtained in the Kestner evaporator by Staub. The overall heat transfer coefficients in the non-boiling section, U_{NB}, obtained in the copper tube are of the same order of those obtained in the boiling sections, U_B, in the steel tube and Kestner evaporators. The value

of/

of U_B/U_{NB} in the Kestner is on an average 1.7, that in the copper tube evaporator is more than 2.

With higher rate of vapour-liquid velocity in the boiling section, a greater overall heat transfer coefficient considering the entire length of the tube can be expected. The velocities of liquid-vapour mixture at the exit of the tube were calculated from the specific volumes of steam and water at the boiling point (taken as 212° F.) and a knowledge of the amount of evaporate collected and the amount of feed. A plot was made of log velocity at exit against the overall heat transfer coefficient, U, considering the entire length of the tube. This is shown in Fig. (30). The value of U falls with the decrease of velocity up to about 45 ft./sec. and then very slowly until it almost reaches a steady value. Vapour velocity at the exit as high as 120 ft. per sec. has been reported by Staub. In the present work the maximum velocity of water-vapour mixture was about 70 ft./sec.

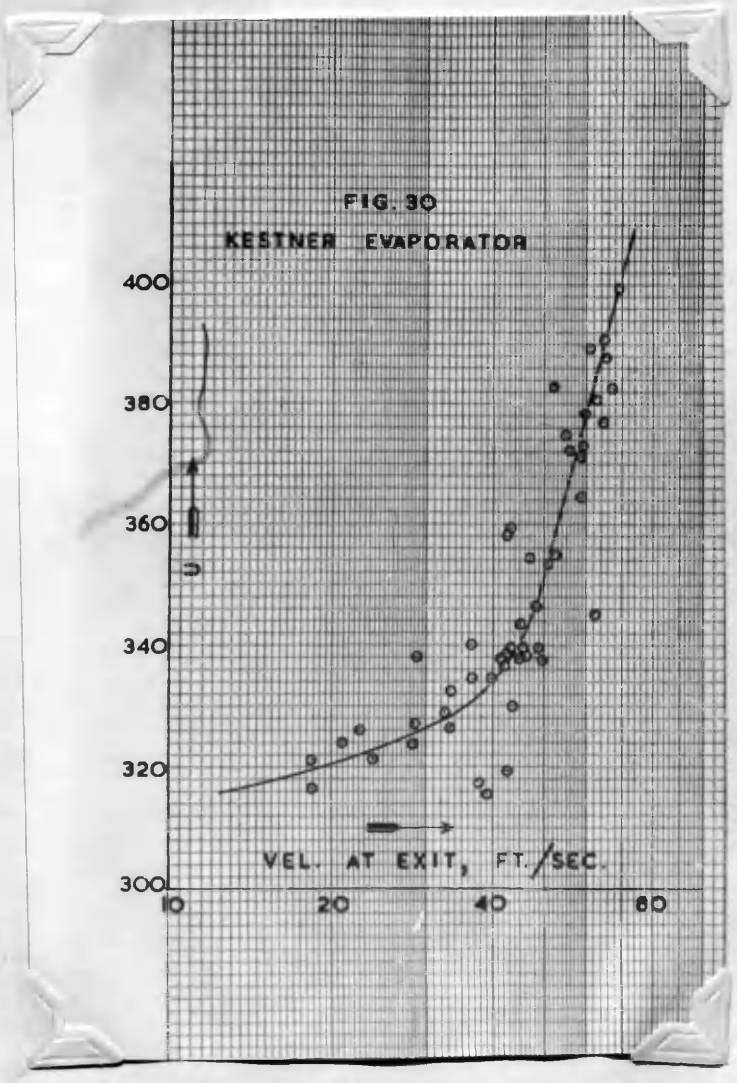


FIG. 30
KESTNER EVAPORATOR
400
380
360
340
320
300
10
20
40
60
VEL. AT EXIT, FT./SEC.

FIG. 30

P A R T II

FORCED CIRCULATION VERTICAL EVAPORATOR.

PART II.

SECTION A.

FORCED CIRCULATION LONG TUBE VERTICAL EVAPORATOR.

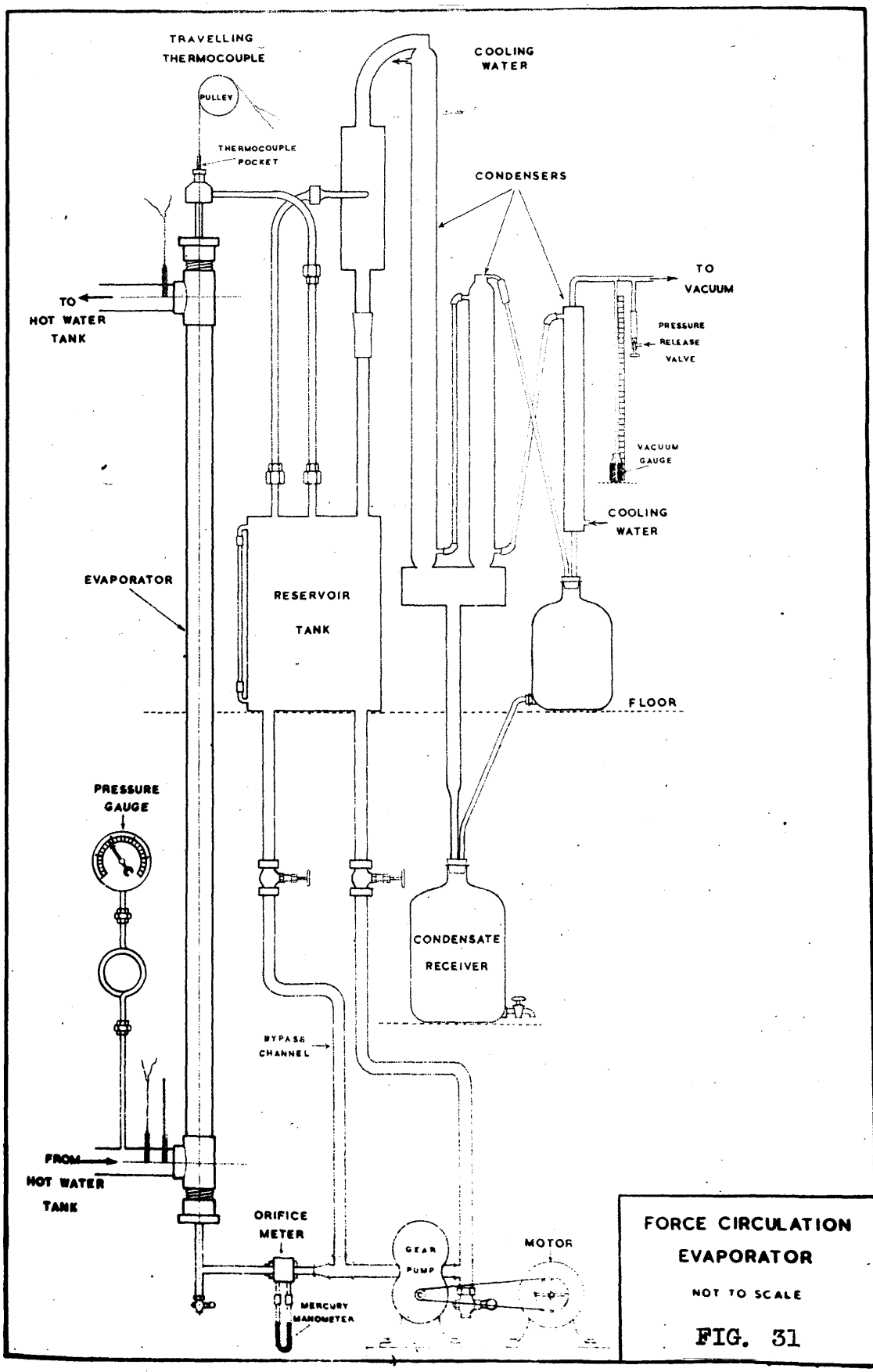
It has been suggested that the liquid film coefficient in the case of the forced-circulation evaporator is much higher than in the case of natural circulation.

A forced-circulation evaporator was designed so that work could be carried out in it. A working diagram of the unit set up, and also a photographic illustration, are shown in Figs. (31 and 32).

APPARATUS:

It was decided that least possible change should be made in this unit from the previous natural circulation unit.

The reservoir tank is a cylindrical copper vessel of about 8 gals. capacity with a liquid level indicator. A $\frac{3}{4}$" copper tube leads from the tank to the inlet of a gear pump which is driven by a $\frac{1}{2}$ H.P. motor. From the outlet of the gear pump comes out a T-piece from one end of which a $\frac{3}{4}$" copper tube enters the bottom of the reservoir tank serving the purpose of a bye-pass channel. From the other end of the T-piece through a reducer is connected an orifice-meter which/



FORCE CIRCULATION EVAPORATOR

NOT TO SCALE

FIG. 31

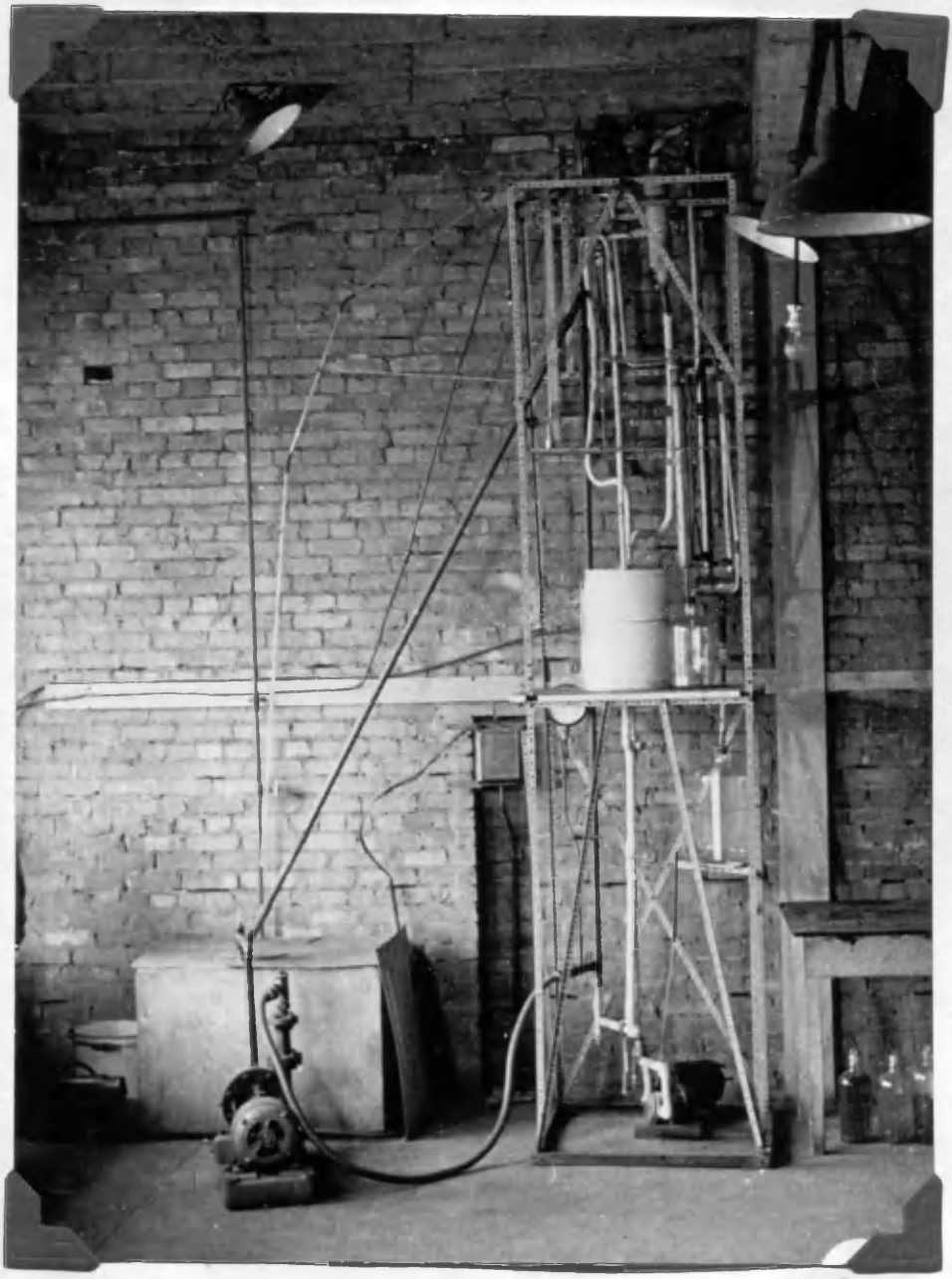
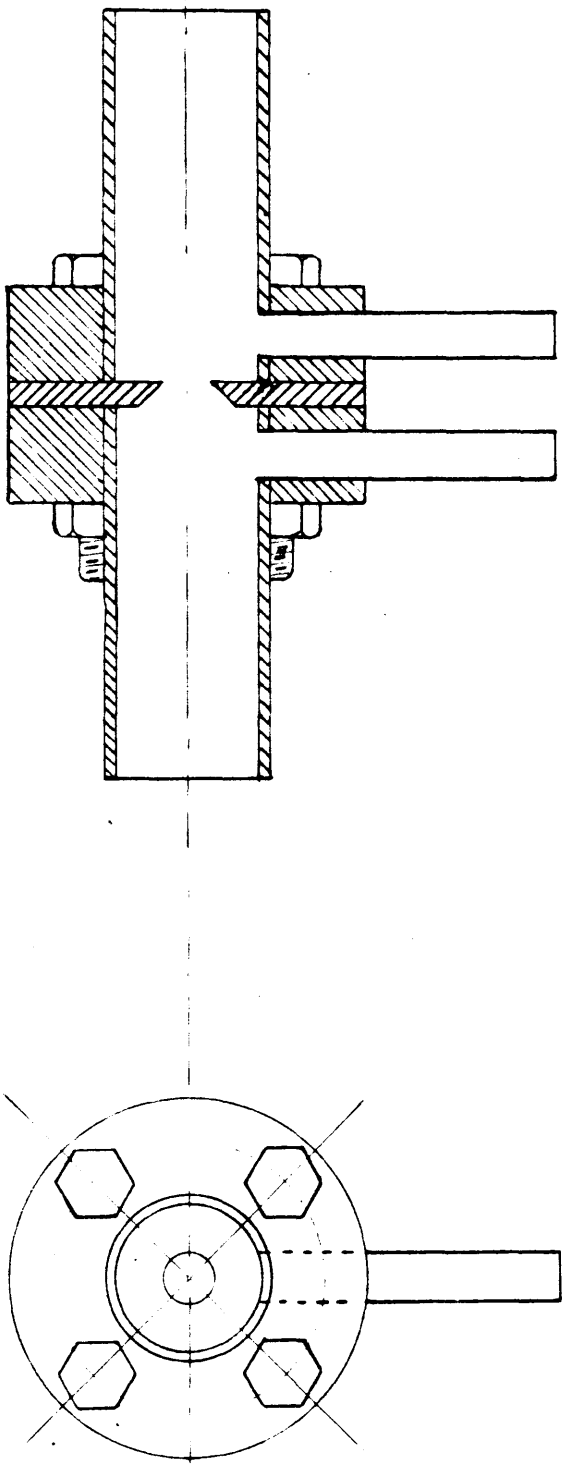


FIG. 32

which in turn is connected with the lower end of the evaporator tube of copper having the same dimensions as used in the natural circulation evaporator. The details of the orifice-meter construction is shown in Fig. (33). The exit of the evaporator tube leads through a ½" copper tube to the top of the reservoir tank. The top of the tank is connected with two copper tubes to the separator and condenser system of the natural circulation unit. Here the reservoir tank serves the purpose of the main liquid vapour separator. The rest of the connection was as used before in the natural circulation evaporator. Though the percentage evaporation is not great, the amount of condensate handled is more due to great rate of circulation. There was the need here of using a third condenser in series with the other two. The whole apparatus was carefully lagged with asbestos cord and sheets so there was very little loss of heat. The reservoir tank could be filled up with the liquid to be experimented with by connecting the small tap at the inlet end of the gear pump and starting the motor.

CALIBRATION OF ORIFICE-METER FOR WATER:

The first thing necessary was to calibrate the orifice meter. The two unions in the tube leading from the top of the/



ORIFICE METER

FULL SCALE

FIG - 33.

the evaporator tube to the reservoir tank greatly facilitated this. By removing the unions, the exit end of the evaporator was connected with a hose pipe which could be switched either to a sink or a weighing tank in a balance. By manipulating the inlet valve to the gear pump and the valve in the bye-pass channel, different flow rates through the orifice were maintained. Two or three readings of the manometer were taken maintaining the same velocity and the mean value was taken. The calibration graph is shown in Fig. (34).

CALIBRATION OF THERMOCOUPLES:

The thermocouples connections were checked up and the travelling thermocouple was calibrated as usual with the fixed thermocouples at 80° C.

(a) LIQUID FILM COEFFICIENT DETERMINATION FOR WATER.

The reservoir tank was filled up with water to two-thirds to three-fourth of its capacity. The gear pump was started and the heating water at 80° C. was circulated through the jacket. It took some time for the water in the reservoir tank to come to the temperature at which it was worked. The temperature reading of the water inside the evaporator tube showed/

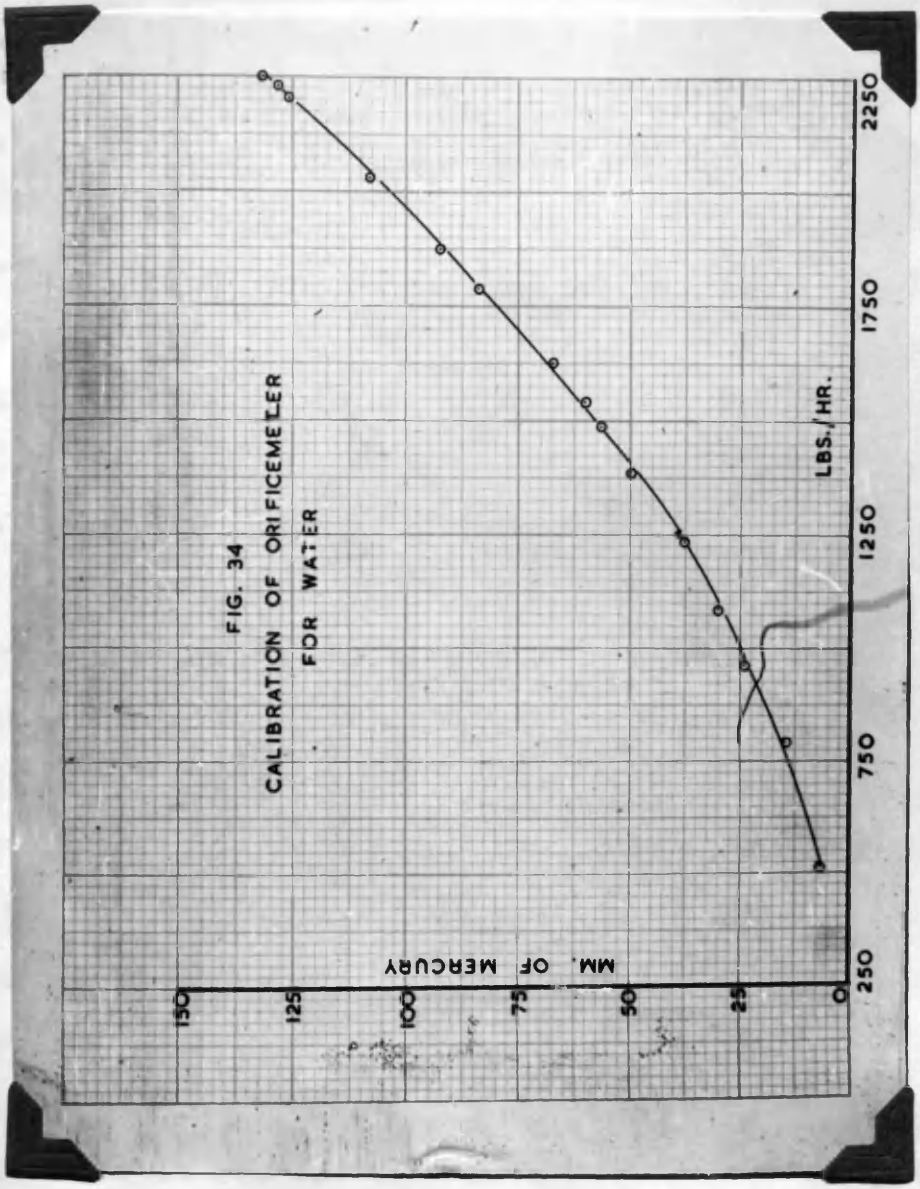


FIG. 34
CALIBRATION OF ORIFICEMETER
FOR WATER
MM. OF MERCURY
150
125
100
75
50
25
0
LBS./HR.
250
750
1250
1750
2250

FIG. 34

showed that it increased gradually as the top of the evaporator tube was reached indicating that there was no boiling taking place in the tube even at the lowest feed rate. After the hot liquid from the evaporator tube reached the reservoir tank, it was relieved of pressure and the evaporation noted was due to the flashing of the liquid. As the liquid entered the evaporator tube almost at the same temperature as existed in the separator (the reservoir tank in this case) and even though no boiling took place in the evaporator tube, evaporation due to flashing was enormous if the system were worked under the same pressure as used in the natural circulation evaporator. This was not feasible under the present circumstances and the pressure under which the system worked had to be increased. In other words the experiments had to be carried out at lower differences of temperatures. In the present set of experiments a vacuum of 55 cms. of mercury was maintained throughout.

Each experiment lasted about 20 minutes during which period thermocouple readings up and down the evaporator tube and the manometer reading in the orifice meter were taken and at the end of the experiment the amount of condensate collected was noted. At the end of each experiment the condensate was transferred from the condensate receiver to the reservoir tank by connecting the tap of the condensate receiver and the tap

at/

at the entrance end of the gear pump with a tube and starting the pump. There was thus no loss of liquid and before starting the next experiment sufficient time was allowed for the steady conditions to be reached. Some 72 runs were made with feed rates from 785 to 2410 lbs. per hour, representing velocities from 3.5 to 11 ft. per second. The method of calculation was the same as used before. The date of results are shown in Table 27.

It will be seen from the table that as the rate of circulation per hour increases, the amount evaporated also increases but the percentage evaporation practically remains the same and is very nearly 1%. It will also be seen that the overall heat transfer coefficient increases with the increased rate of circulation. In the table (27), the velocity calculated is the velocity of the liquid at the exit of the evaporator tube. As no boiling took place in the evaporator tube, no consideration was given to any vapour, if any, in this calculation. When the log of velocity at exit is plotted against the overall heat transfer coefficient, almost all the points can be represented by a straight line as shown in the Fig. (35). It can be easily seen that the plot of log V against the liquid film heat transfer coefficient, h, will be of the same nature.

(b)/

TABLE - 27

WATER IN FORCED CIRCULATION COPPER TUBE EVAPORATOR.

Heating Water at 80° C.

Vacuum - 55 cms. of Mercury.

No Boiling Occured in the Tube.

Test No.	Feed Rate lbs/hr.	Temp. in Separator. °F	Evapn. lbs/hr.	% Evapn.	Q B.Th.U/hr.	ΔT_m °F	U B.Th.U./hr.	h B.Th.U./hr.	Velocity at exit. Ft./sec.
1.	785	158.4	7.73	0.99	7770	11.38	535	681	3.560
2.	825	159.1	7.94	0.96	7980	11.15	562	724	3.744
3.	930	159.0	9.86	1.06	9910	11.46	678	930	4.220
4.	962	155.7	11.64	1.21	11700	13.01	705	987	4.368
5.	990	158.5	10.50	1.06	10550	11.90	696	966	4.490
6.	992	155.2	12.89	1.30	12960	13.49	754	1079	4.500
7.	1055	156.4	12.03	1.14	12090	12.75	744	1059	4.790
8.	1075	157.9	10.92	1.02	10975	12.35	698	968	4.880
9.	1101	156.6	12.90	1.17	12970	12.87	790	1155	4.996
10.	1138	155.0	14.52	1.28	14620	14.12	812	1203	5.165
11.	1165	159.0	11.98	1.03	12050	11.82	799	1174	5.286
12.	1188	158.5	11.90	1.00	11960	11.84	793	1160	5.394
13.	1195	159.0	12.24	1.02	12305	11.92	810	1190	5.420
14.	1293	154.4	16.20	1.25	16300	14.35	891	1384	5.870
15.	1310	159.4	12.11	0.93	12170	11.79	810	1190	5.950

Continued/-

TABLE - 27 (Continued)

Test No.	Feed Rate lbs/hr.	Temp. in Separaotr °F	Evapn. lbs/ hr.	% Evapn.	Q B.Th.U/hr.	ΔT_m °F	U B.Th.U./hr.	h B.Th.U./hr.	Velocity at exit. Ft./sec.
16.	1385	159.0	13.23	0.96	13300	11.96	873	1340	6,282
17.	1400	158.2	13.70	0.98	13770	12.50	865	1322	6.350
18.	1410	155.5	16.54	1.17	16620	14.08	926	1471	6.400
19.	1467	155.0	17.87	1.22	17980	14.35	983	1620	6.660
20.	1475	157.5	15.22	1.03	15300	12.98	924	1466	6.692
21.	1480	159.1	14.21	0.96	14280	12.24	917	1447	6.720
22.	1485	155.0	17.87	1.20	17980	14.29	987	1631	6.740
23.	1500	160.4	13.29	0.89	13355	11.58	904	1412	6.806
24.	1537	157.8	15.88	1.03	15960	12.87	974	1595	6.972
25.	1540	155.2	18.51	1.20	18620	14.19	1029	1749	6.990
26.	1580	159.1	14.44	0.91	14520	12.25	930	1481	7.167
27.	1599	156.6	17.74	1.11	17880	13.49	1036	1770	7.250
28.	1600	155.7	18.65	1.17	18750	14.42	1020	1723	7.350
29.	1630	156.6	16.58	1.02	16660	13.72	953	1540	7.392
30.	1650	157.3	16.93	1.03	17020	13.22	1010	1695	7.490
31.	1700	159.4	15.35	0.90	15430	12.14	998	1660	7.715
32.	1701	157.3	17.20	1.01	17290	13.40	1012	1701	7.718
33.	1710	159.0	15.16	0.89	15240	12.60	948	1527	7.767
34.	1711	155.5	19.85	1.16	19950	14.54	1076	1888	7.767

Continued/-

TABLE - 27 (Continued)

Test No.	Feed Rate lbs/hr	Temp. in Separator °F	Evapn. lbs/hr.	% Evapn.	Q B.Th.U./hr.	ΔT_m °F	U B.Th.U./hr.	h B.Th.U./hr.	Velocity at exit. Ft./sec.
3 5	1717	158.0	17.38	1.02	17470	13.10	1046	1799	7.790
36.	1737	157.0	18.52	1.07	18600	13.72	1064	1852	7.880
37.	1790	156.8	17.86	1.00	17950	13.42	1050	1810	8.120
38.	1795	155.7	20.16	1.12	20260	14.54	1093	1942	8.145
39.	1796	156.3	18.96	1.06	19060	13.84	1081	1906	8.150
40.	1800	159.4	14.39	0.80	14460	11.70	970	1585	8.166
41.	1850	157.9	17.90	0.97	17990	13.12	1076	1890	8.392
42.	1853	157.5	18.98	1.02	19060	13.49	1109	1965	8.410
43.	1860	158.0	18.08	0.97	18170	13.18	1082	1901	8.445
44.	1890	159.4	16.67	0.88	16760	12.27	1072	1876	8.580
45.	1916	157.3	19.85	1.03	19940	13.76	1137	2085	8.692
46.	1940	159.1	16.67	0.86	16760	12.62	1041	1784	8.800
47.	1945	157.6	18.89	0.97	18990	13.43	1109	1993	8.822
48.	1960	157.6	19.13	0.98	19230	13.33	1131	2066	8.900
49.	1980	157.5	18.75	0.95	18850	13.38	1105	1980	8.980
50.	1981	157.9	18.80	0.95	18895	13.53	1095	1949	8.990
51.	1982	157.0	20.16	1.02	20250	13.86	1147	2120	9.000
52.	1990	159.1	17.65	0.89	17740	12.47	1117	2021	9.030
53.	2000	158.9	20.98	1.05	21100	13.00	1273	2594	9.074

Continued /-

TABLE - 27 (Continued)

Test No.	Feed Rate lbs/hr.	Temp. in Separator °F	Evapn. lbs/hr.	% Evapn.	Q B.Th.U./hr.	ΔT_m °F	U B.Th.U./hr.	h B.Th.U./hr.	Velocity at exit. Ft./sec.
54.	2028	156.1	21.43	1.06	21540	14.40	1173	2210	9.208
55.	2048	157.7	19.65	0.96	19730	13.81	1120	2029	9.300
56.	2065	157.0	22.05	1.07	22150	13.93	1248	2492	9.370
57.	2086	157.8	19.84	0.95	19930	13.41	1166	2185	9.470
58.	2106	157.8	19.49	0.93	19580	13.38	1148	2127	9.555
59.	2125	157.6	20.10	0.95	20200	13.35	1188	2263	9.645
60.	2135	157.6	20.28	0.95	20390	13.45	1190	2269	9.690
61.	2165	158.0	19.94	0.92	20050	13.28	1185	2253	9.826
62.	2175	159.1	17.65	0.81	17740	12.61	1104	1976	9.872
63.	2213	158.4	19.93	0.90	20030	13.14	1196	2293	10.04
64.	2223	157.6	20.28	0.91	20380	14.82	1080	1902	10.08
65.	2252	158.0	19.94	0.89	20050	13.26	1187	2260	10.22
66.	2280	156.4	25.17	1.12	25300	15.40	1290	2665	10.35
67.	2282	158.2	23.99	1.05	24070	13.58	1390	3131	10.36
68.	2300	160.4	17.90	0.78	17990	11.65	1212	2352	10.44
69.	2322	156.3	22.75	0.98	22860	14.66	1224	2398	10.54
70.	2325	155.7	23.14	0.99	23250	14.73	1238	2452	10.55
71.	2340	159.3	18.52	0.79	18620	12.39	1179	2229	10.62
72.	2410	156.6	23.94	0.99	24040	14.62	1290	2665	10.99

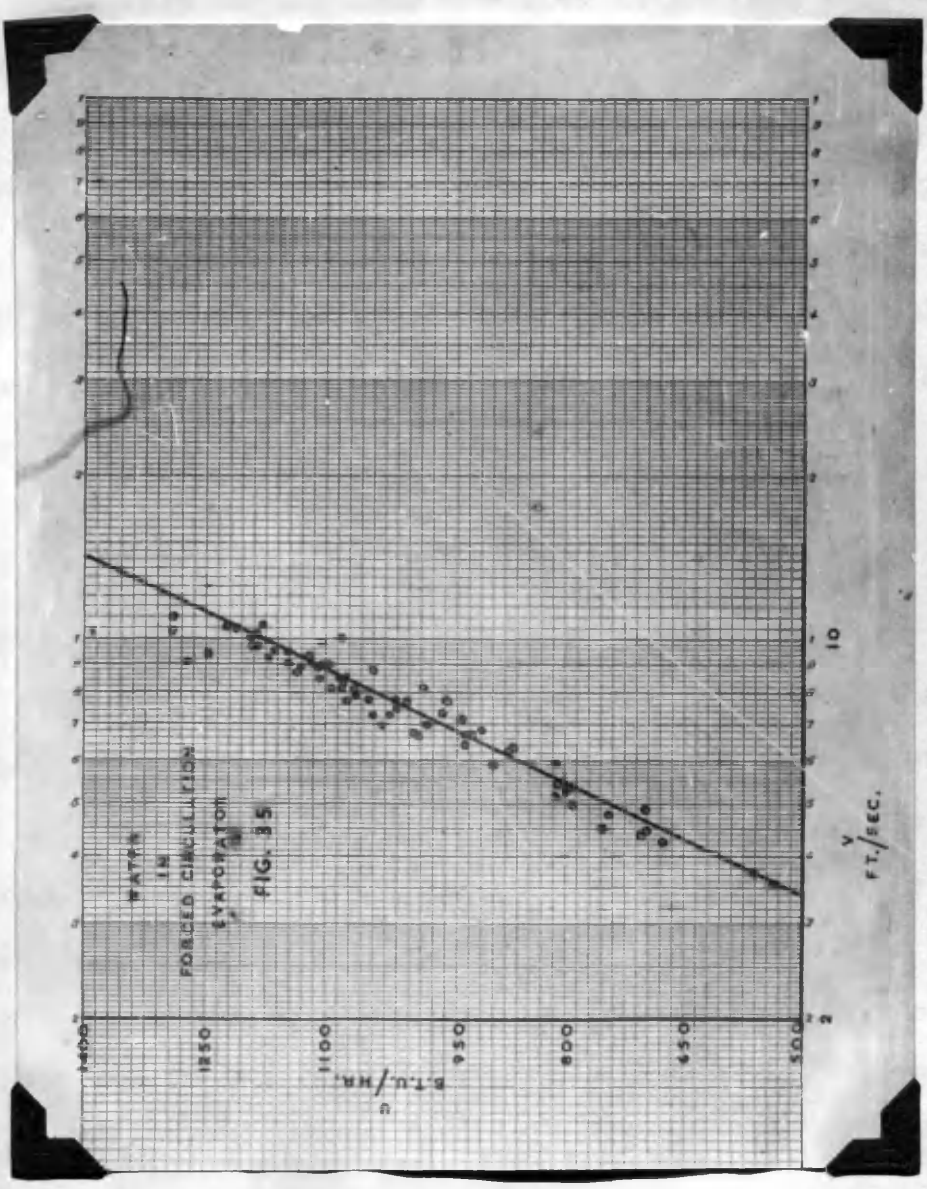

WATER
IN
FORCED CIRCULATION
EVAPORATOR
FIG. 35
B.T.U./HR.
1250
1100
950
800
650
500
2
10
V
FT./SEC.

FIG. 35

(b) CORRELATION OF DATA.

The data on the heat transfer to boiling liquids in forced circulation evaporators are very few. The following two papers are of interest. Logan, Fragen and Badger (41) working in a semi-commercial forced circulation evaporator (8 ft. tubes, $\frac{3}{4}$" I.D., and 18.4 sq.ft. of heating surface) with sugar solutions of different concentrations under essentially non-boiling conditions, found that the Dittus-Boelter equation

$$hD/K = a\ (DV\rho/\mu)^{0.8}\ (\mu C/K)^{0.4}$$

correlated their data very well.

Boarts, Badger and Meisenburg (9) worked on a single tube (12 ft. long, 0.76" I.D. with inside heating surface of 2.367 sq.ft.) forced circulation evaporator with distilled water. They had some boiling in the tubes but the Dittus-Boelter equation correlated their data also though their value of the constants was a little higher than the figures reported by Logan et al. In the present correlation the same step was taken.

The Prandtl group ($C\mu/K$) was not a primary variable in this work. Therefore, the experience of the past workers in/

in the general field was drawn upon, and this group was introduced raised to the 0.4 power.

Velocity was calculated at the average temperature of the liquid in the tube and so also the values of μ, C, K and ρ from the appropriate tables.

The Reynold and Nusselt numbers, $(DV\rho/\mu)$ and hD/K were calculated from the observed data (Table 28). In these calculations consistent units were used throughout. To find the power function of the Reynold's number a plot was made of

$$\frac{hD/K}{(\mu C/K)^{0.4}} \quad \text{against} \quad (DV\rho/\mu)$$

This is shown in Fig. (36). It will be observed that a line with a slope of 0.9 on the logarithmic plot best correlates the data. The value of the constant was then calculated from

$$a = \frac{hD/K}{(\mu C/K)^{0.4}\,(DV\rho/\mu)^{0.9}}$$

The average value of "a" was 0.00532. So the equation stands as

$$hD/K = 0.00532\,(DV\rho/\mu)^{0.9}\,(\mu C/K)^{0.4}$$

for/

TABLE - 28

WATER IN FORCED CIRCULATION EVAPORATOR.

Test No.	$\frac{\mu C}{K}$	Vel. at av. temp. ft/sec. V	$\frac{DV\rho}{\mu}$	$\frac{hD}{K}$ Obs.	$\frac{\frac{hD}{K}}{\left(\frac{\mu C}{K}\right)^{0.4}}$	a Cal.
1.	2.424	3.555	32580	67.8	47.6	.00413
2.	2.424	3.736	34300	72.1	50.6	.00420
3.	2.424	4.213	38550	92.6	65.0	.00485
4.	2.424	4.358	39500	98.5	69.1	.00504
5.	2.424	4.482	40900	96.3	67.6	.00478
6.	2.424	4.490	40530	107.7	75.6	.00535
7.	2.424	4.781	43410	105.6	74.1	.00497
8.	2.424	4.870	44310	96.5	67.4	.00444
9.	2.424	4.986	45200	115.2	80.8	.00522
10.	2.423	5.152	46260	120.1	84.3	.00534
11.	2.424	5.276	48280	116.9	82.0	.00500
12.	2.424	5.380	49200	115.5	81.0	.00485
13.	2.424	5.410	49400	118.5	83.1	.00496
14.	2.424	5.852	52200	138.3	97.0	.00551
15.	2.424	5.930	54240	118.5	83.1	.00456
16.	2.424	6.270	57200	133.4	93.6	.00490

Continued /-

TABLE - 28 (Continued)

Test No.	$\frac{\mu C}{K}$	Vel. at av. temp. ft/sec. V	$\frac{DV\rho}{\mu}$	$\frac{hD}{K}$ Obs.	$\frac{\frac{hD}{K}}{\left(\frac{\mu C}{K}\right)^{0.4}}$	a Cal.
17.	2.424	6.340	57680	131.8	92.5	0.00480
18.	2.423	6.380	57350	146.9	103.1	0.00538
19.	2.423	6.643	59600	161.9	113.6	0.00572
20.	2.424	6.680	60560	146.2	102.6	0.00509
21.	2.424	6.702	61100	144.2	101.2	0.00499
22.	2.423	6.725	60400	163.0	114.4	0.00569
23.	2.424	6.794	62200	140.6	98.6	0.00478
24.	2.424	6.960	63100	159.0	111.5	0.00534
25.	2.423	6.972	62650	174.7	122.6	0.00590
26.	2.423	7.150	65200	148.0	103.8	0.00483
27.	2.424	7.240	65370	176.4	123.8	0.00574
28.	2.423	7.248	65080	172.2	120.8	0.00562
29.	2.423	7.380	66500	153.8	107.9	0.00493
30.	2.424	7.475	67620	169.0	118.6	0.00534
31.	2.424	7.700	70250	165.3	116.0	0.00504
32.	2.424	7.706	69560	169.6	119.0	0.00521
33.	2.424	7.745	70400	152.2	106.8	0.00462
34.	2.423	7.750	69450	188.7	132.4	0.00581
35.	2.424	7.776	70390	179.4	125.9	0.00546

Continued /-

TABLE - 28 (Continued)

Test No.	$\frac{\mu C}{K}$	Vel. at av. temp. ft./sec. V	$\frac{DV\rho}{\mu}$	$\frac{hD}{K}$ Obs.	$\frac{\frac{hD}{K}}{\left(\frac{\mu C}{K}\right)^{0.4}}$	a cal.
36.	2.423	7.870	70930	184.8	129.6	0.00558
37.	2.424	8.105	73100	180.6	126.7	0.00531
38.	2.423	8.130	72900	194.1	136.2	0.00572
39.	2.423	8.130	73200	190.2	133.4	0.00558
40.	2.424	8.150	75250	157.8	110.7	0.00452
41.	2.424	8.375	75800	188.4	132.2	0.00537
42.	2.424	8.390	75700	196.0	137.6	0.00560
43.	2.424	8.420	76270	189.6	133.0	0.00537
44.	2.424	8.562	77900	186.8	131.0	0.00519
45.	2.423	8.680	78200	208.1	146.0	0.00576
46.	2.424	8.785	79150	177.8	124.8	0.00487
47.	2.424	8.810	79530	198.8	140.2	0.00545
48.	2.424	8.875	80140	206.0	144.6	0.00558
49.	2.424	8.965	80950	197.4	138.4	0.00529
50.	2.424	8.970	80960	194.4	136.4	0.00522
51.	2.423	8.975	80850	211.5	148.4	0.00568
52.	2.424	9.010	81950	201.6	141.4	0.00535
53.	2.424	9.060	82100	258.8	181.6	0.00686
54.	2.423	9.190	82400	210.9	148.0	0.00557

Continued /-

TABLE - 28 (Continued)

Test No.	$\frac{\mu C}{K}$	Vel. at av. temp. ft./sec. V	$\frac{DV\rho}{\mu}$	$\frac{hD}{K}$ Obs.	$\frac{\frac{hD}{K}}{\left(\frac{\mu C}{K}\right)^{0.4}}$	a Cal.
55.	2.423	9.275	83560	202.3	142.0	0.00528
56.	2.423	9.355	84200	248.6	174.4	0.00643
57.	2.424	9.450	85300	218.0	152.9	0.00558
58.	2.424	9.540	86100	212.2	148.8	0.00539
59.	2.424	9.625	86950	225.8	158.5	0.00568
60.	2.424	9.670	87360	226.4	158.9	0.00567
61.	2.424	9.800	88550	189.6	133.0	0.00470
62.	2.424	9.850	89400	197.0	138.3	0.00484
63.	2.424	10.020	90800	228.6	160.4	0.00553
64.	2.423	10.070	90000	190.1	133.4	0.00464
65.	2.424	10.200	92200	225.4	158.2	0.00538
66.	2.423	10.330	91960	266.6	187.0	0.00638
67.	2.424	10.340	93300	312.6	219.4	0.00738
68.	2.424	10.420	95200	234.0	164.2	0.00543
69.	2.423	10.520	94100	239.7	168.2	0.00562
70.	2.423	10.530	94150	245.1	172.0	0.00575
71.	2.424	10.600	96400	222.0	155.8	0.00509
72.	2.423	10.920	97700	266.4	186.9	0.00604
					Av.	0.00532

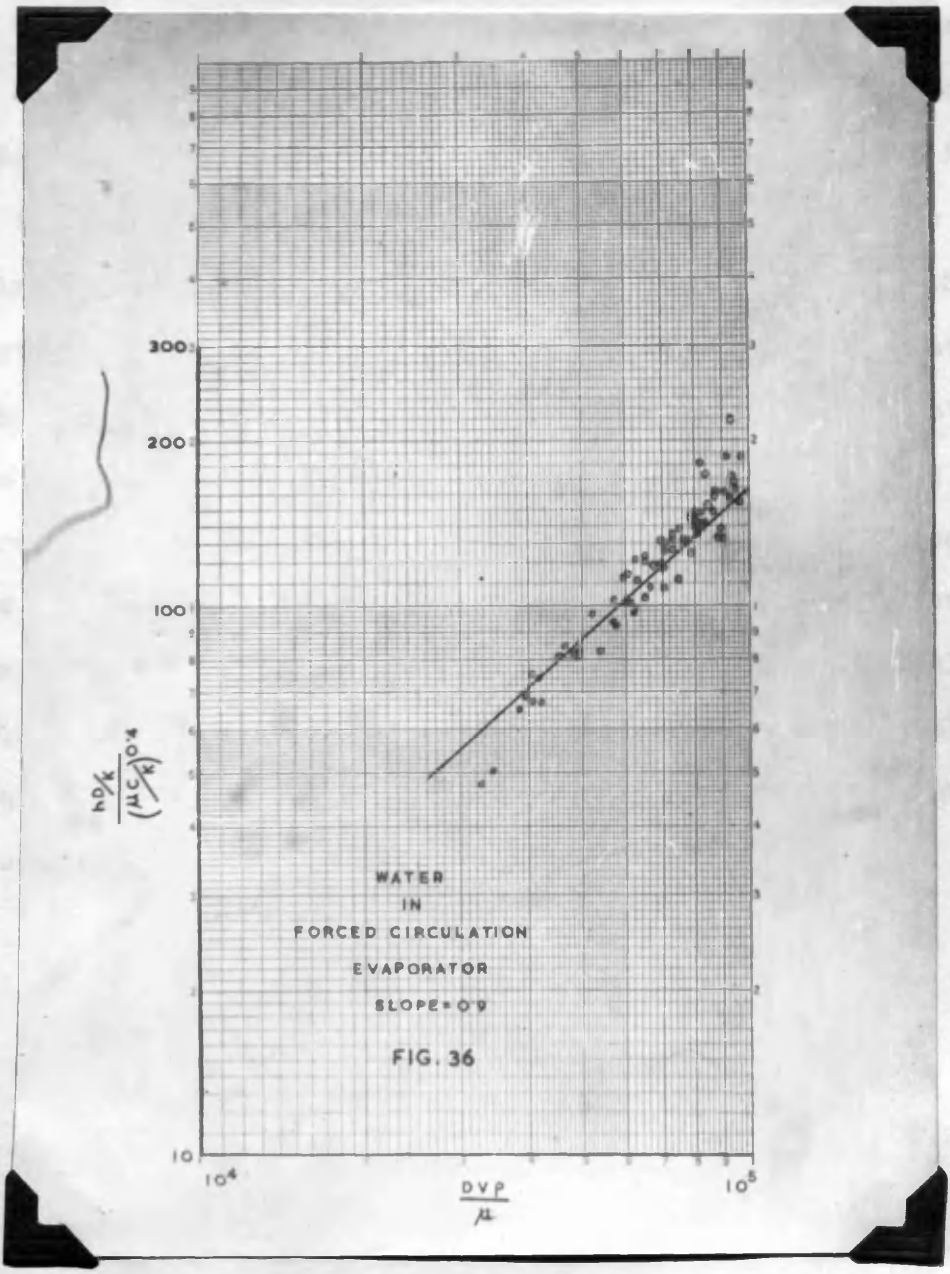

300
200
100
10
10^4
10^5
$\frac{DVP}{\mu}$
$\frac{hD/k}{(\mu C/k)^{0.4}}$
WATER
IN
FORCED CIRCULATION
EVAPORATOR
SLOPE = 0.9
FIG. 36

FIG. 36

for the best correlation of the observed values of "h".

It will be observed that the value of the constant "a" is almost one-fourth of the value of 0.0225 suggested by McAdams (42).The value of "a" given by Boarts et al is 0.0278. In the present work the method of calculation of the liquid film coefficient was quite different from those of the workers mentioned above. Boarts et al measured the temperature drop across the liquid film by measuring the temperature of the tube wall by thermocouples and these values were used directly for finding out the liquid film coefficients. In the present work the overall heat transfer coefficients were first found out and with a knowledge of the previously determined value of the combined annular film and wall coefficients, the liquid film coefficients were calculated from the relation

$$\frac{1}{U} - \left(\frac{1}{H_A} + \frac{1}{H_W}\right) = \frac{1}{h}$$

PART II.

SECTION B.

(a) LIQUID FILM COEFFICIENT FOR ALCOHOL.

Work was then taken up with organic liquids. First ethyl alcohol was used as feed liquid.

The orifice meter was first calibrated for alcohol. The two unions, joining the exit end of the evaporator tube and the top of the reservoir tank, were opened and the exit end of the evaporator tube was connected with a rubber tube which could be switched either to the reservoir tank or a calibrated aspirator bottle. The unit was dried and then washed with alcohol. The tank was then filled with alcohol to about two-thirds the capacity and the gear pump started. When no reading was being taken the alcohol was only being circulated from the tank and back to it. While taking a reading, the rubber tube was switched on into a calibrated aspirator bottle and the volume of alcohol collected for a known period of time was noted together with the manometer reading of the orifice meter. The volume of alcohol was then converted into pounds per hour. The calibration curve is shown in Fig. 37.

The/

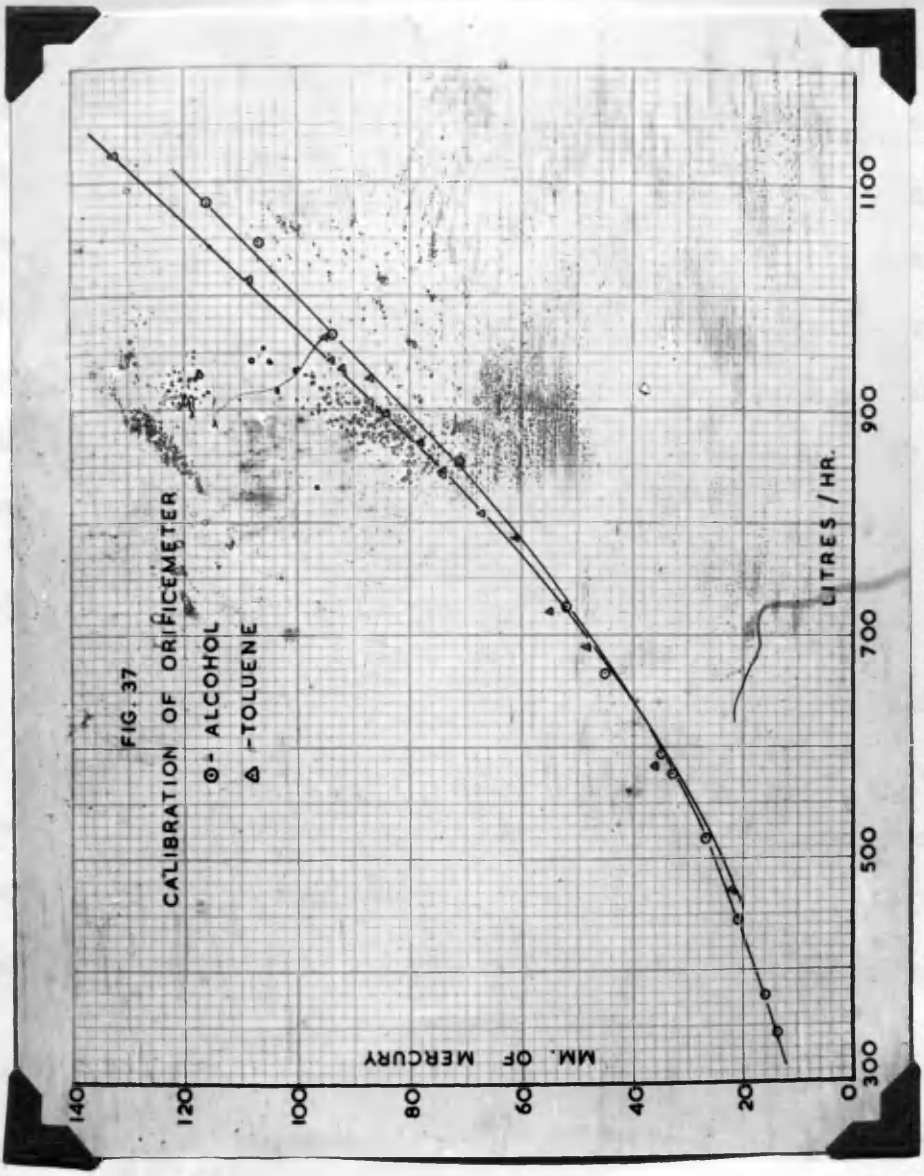


FIG. 37
CALIBRATION OF ORIFICEMETER
⊙- ALCOHOL
△ -TOLUENE
MM. OF MERCURY
LITRES / HR.
0
20
40
60
80
100
120
140
300
500
700
900
1100

FIG. 37

The travelling thermocouple was calibrated with the fixed thermocouples at the inlet and outlet of heating water at 50° C. Experiments were carried out in the same way as in the case of water. Some 45 runs were made with feed rates varying from 841 to 1949 lbs./hr. representing velocities from 4.8 ft./sec. to 11.2 ft./sec. A vacuum of 62 cms. of mercury was maintained throughout the experiments. Temperatures readings of the liquid inside the evaporator tube showed that no boiling took place in any of the runs. The calculations were done in the same way as used before. The data are given in Table 29.

(b) CORRELATIONS.

For the correlation, the same principle was followed as in the correlation of water. All the necessary values of the physical properties of alcohol were taken from reliable tables and nomographs. All the necessary values calculated from the observed data are shown in Table 30. A plot of

$$\frac{hD/K}{(\mu C/K)^{0.4}} \qquad \text{against} \qquad (DV\rho/\mu)$$

shows (Fig. 38) that the power function of Reynolds number has /-

TABLE - 29

ALCOHOL IN FORCED CIRCULATION COPPER TUBE EVAPORATOR.

Heating Water at - 50° C.

Vacuum - 62 cms. of mercury.

No Boiling in any test.

Test No.	Feed Rate litres /hr.	Feed Rate lbs/ hr.	Evapn. lbs/hr.	% Evapn.	Temp. in Separa-tor. °f	Q B.Th.U./hr.	ΔT_m °F	U B.Th.U./hr.	h B.Th.U./hr.
1.	494	841	7.48	0.890	106.7	2842	11.79	189.1	206.7
2.	574	977	8.36	0.856	106.5	3176	11.36	219.3	243.3
3.	575	979	9.86	1.007	105.2	3745	12.49	235.2	298.3
4.	600	1021	9.45	0.925	105.6	3591	11.74	240.0	269.0
5.	650	1107	10.96	1.078	105.6	4165	12.42	263.0	298.3
6.	652	1110	11.12	1.002	105.3	4225	12.86	257.5	291.3
7.	665	1131	10.43	0.922	106.3	3965	11.92	260.4	295.0
8.	680	1157	10.78	0.931	106.0	4095	11.93	269.2	306.3
9.	695	1183	10.90	0.921	105.0	4142	12.47	260.5	295.0
10.	767	1305	12.18	0.933	105.8	4627	12.74	28.50	327.0
11.	794	1352	12.57	0.930	105.3	4777	12.24	306.0	354.9
12.	801	1363	13.22	0.970	105.8	5025	12.60	312.8	364.0
13.	820	1396	12.88	0.923	105.6	4897	12.77	300.6	347.6

Continued /-

TABLE - 29 (Continued)

Test No.	Feed Rate litres /hr.	Feed Rate lbs. /hr.	Evapn. lbs/hr.	% Evapn.	Temp. in Separator °f	Q B.Th.U./hr.	ΔT_m °F	U B.Th.U./hr.	h B.Th.U./hr.
14.	826	1406	13.98	0.995	106.3	5311	12.28	339.4	400.6
15.	865	1472	13.84	0.940	106.0	5260	12.07	341.8	403.9
16.	870	1481	14.53	0.982	105.3	5525	12.74	340.3	401.8
17.	872	1484	15.05	1.014	105.3	5717	12.96	346.0	409.8
18.	895	1524	14.26	0.936	105.6	5420	12.09	351.5	417.5
19.	900	1532	14.40	0.940	105.8	5471	12.55	342.2	404.5
20.	931	1584	13.22	0.834	107.3	5025	11.38	346.5	410.5
21.	933	1587	13.84	0.872	106.0	5260	12.07	341.8	403.9
22.	934	1589	15.22	0.958	105.6	5781	12.64	358.7	427.7
23.	936	1592	14.61	0.918	106.5	5555	12.05	361.7	432.0
24.	943	1604	15.60	0.972	105.8	5940	12.87	362.0	432.5
25.	958	1630	15.43	0.946	105.6	5863	12.73	361.3	432.6
26.	965	1642	14.27	0.869	107.2	5425	11.60	366.6	439.0
27.	990	1684	15.71	0.934	105.6	5970	12.34	379.6	458.2
28.	996	1695	15.64	0.923	106.5	5940	12.24	381.0	460.0
29.	999	1700	15.22	0.895	107.1	5785	11.75	386.0	467.1

Continued /-

TABLE - 29 (Continued)

Test No.	Feed Rate litres /hr.	Feed Rate lbs. /hr.	Evapn. lbs. /hr.	% Evapn.	Temp. in Separator. °F	Q B.Th.U./hr.	ΔT_m °F.	U B.Th.U./hr.	h B.Th.U./hr.
30.	1010	1719	15.22	0.885	106.0	5785	12.32	368.0	441.1
31.	1015	1727	16.61	0.963	106.3	6312	12.67	391.0	474.5
32.	1025	1744	15.22	0.873	106.5	5783	11.59	39.15	475.2
33.	1027	1748	15.91	0.911	106.7	6045	12.03	394.3	479.3
34.	1040	1770	16.34	0.923	106.2	6211	12.57	387.8	469.8
35.	1055	1794	16.61	0.926	105.9	6312	12.40	399.2	486.4
36.	1060	1803	16.08	0.892	105.4	6113	12.40	387.0	468.6
37.	1065	1811	16.13	0.891	106.5	6132	12.06	398.5	485.6
38.	1080	1837	16.61	0.904	107.1	6312	11.83	418.5	515.6
39.	1095	1863	16.68	0.895	106.3	6337	12.15	409.0	501.3
40.	1116	1899	16.41	0.865	105.3	6235	12.01	407.5	499.0
41.	1120	1906	16.74	0.879	106.7	6360	12.03	411.0	504.3
42.	1130	1924	15.79	0.821	106.7	6001	11.25	418.5	515.6
43.	1132	1927	17.45	0.906	106.9	6631	11.99	434.0	539.3
44.	1136	1934	16.88	0.873	106.3	6412	11.98	420.0	517.9
45.	1145	1949	16.62	0.853	107.1	6320	11.75	422.0	519.4

T A B L E - 30

ALCOHOL IN FORCED CIRCULATION EVAPORATOR.

Test No.	Velocity ft./sec. V	$\frac{DV\rho}{\mu}$	$\frac{hD}{K}$ Obs.	$\frac{hD/K}{\left(\mu C/K\right)^{0.4}}$	a Cal.
1.	4.82	16420	89.6	30.80	0.00506
2.	5.60	19090	105.4	36.28	0.00509
3.	5.61	19130	129.3	44.50	0.00637
4.	5.85	19950	116.6	40.12	0.00541
5.	6.34	21640	129.3	44.50	0.00558
6.	6.36	21700	126.3	43.50	0.00544
7.	6.48	22100	127.9	44.02	0.00532
8.	6.63	22600	132.8	45.70	0.00551
9.	6.78	23100	127.9	44.02	0.00520
10.	7.47	25450	141.8	48.80	0.00529
11.	7.74	26380	153.9	52.95	0.00556
12.	7.81	26630	157.9	54.34	0.00566
13.	8.00	27290	150.7	51.85	0.00528
14.	8.06	27490	173.7	59.76	0.00604
15.	8.43	28730	173.2	60.30	0.00586
16.	8.48	28900	174.2	59.92	0.00579
17.	8.50	28980	177.7	61.20	0.00600
18.	8.73	29750	181.0	62.30	0.00587
19.	8.78	29930	175.4	60.34	0.00566
20.	9.08	30960	178.0	61.30	0.00557

Continued /-

TABLE - 30 (Continued)

Test NO.	Vel. ft/sec. V	$\frac{DV\rho}{\mu}$	$\frac{hD}{K}$ Obs.	$\frac{\frac{hD}{K}}{\left(\frac{\mu C}{K}\right)^{0.4}}$	a Cal.
21.	9.10	31040	175.2	60.30	0.00547
22.	9.11	31070	185.3	63.80	0.00578
23.	9.12	31100	187.2	64.42	0.00578
24.	9.19	31350	187.4	64.50	0.00580
25.	9.34	31840	187.5	64.55	0.00572
26.	9.41	32070	190.3	65.50	0.00577
27.	9.65	32900	198.7	68.40	0.00588
28.	9.71	33100	199.4	68.65	0.00587
29.	9.74	33200	202.5	69.70	0.00595
30.	9.85	33590	191.3	65.80	0.00543
31.	9.90	33750	205.7	70.80	0.00582
32.	9.99	34070	205.9	70.90	0.00591
33.	10.00	34100	207.6	71.50	0.00596
34.	10.15	34600	203.5	70.05	0.00576
35.	10.30	35120	210.8	72.56	0.00588
36.	10.34	35260	203.0	69.90	0.00565
37.	10.39	35420	210.4	72.40	0.00583
38.	10.54	35930	223.5	76.95	0.00611
39.	10.68	36400	217.3	74.80	0.00588
40.	10.89	37100	216.3	74.42	0.00575
41.	10.93	37250	218.5	75.20	0.00579
42.	11.03	37600	223.5	76.95	0.00588
43.	11.05	37660	233.6	80.40	0.00612
44.	11.09	37800	224.5	77.30	0.00587
45.	11.17	38060	225.2	77.50	0.00585
				Av. -	0.00571

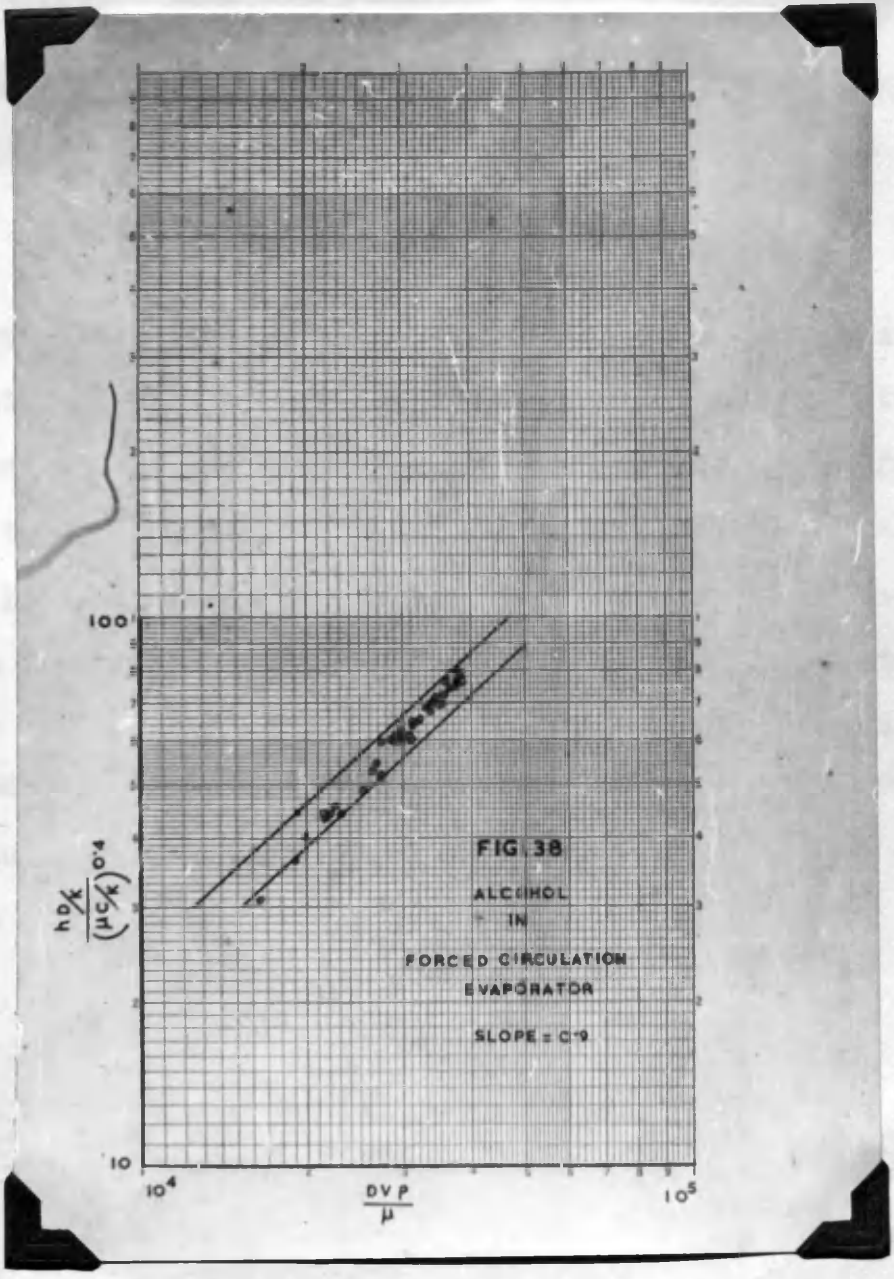

100
10
10^4
10^5
$\frac{hD}{k}\left(\frac{\mu c_p}{k}\right)^{0.4}$
$\frac{DVP}{\mu}$
FIG 38
ALCOHOL
IN
FORCED CIRCULATION
EVAPORATOR
SLOPE = C·9

FIG. 38

has a slope of 0.9. The equation which satisfies the observed film coefficients is

$$\frac{hD}{K} = 0.00571 \left(\frac{DV\rho}{\mu}\right)^{0.9} \left(\frac{\mu C}{K}\right)^{0.4}$$

The constant is a little higher than that of the equation for water. However, taking an average value of the two constants as 0.0055 , a single equation can be used for the calculation of film coefficients for both water and alcohol.

In the case of water it was found that the velocities of the liquid at the exit and at the average temperatures of the liquid were almost the same. So in this case a plot was made of log velocity against the overall heat transfer coefficient, Fig. 39. It has the same nature as that shown in the case of water in Fig. 35.

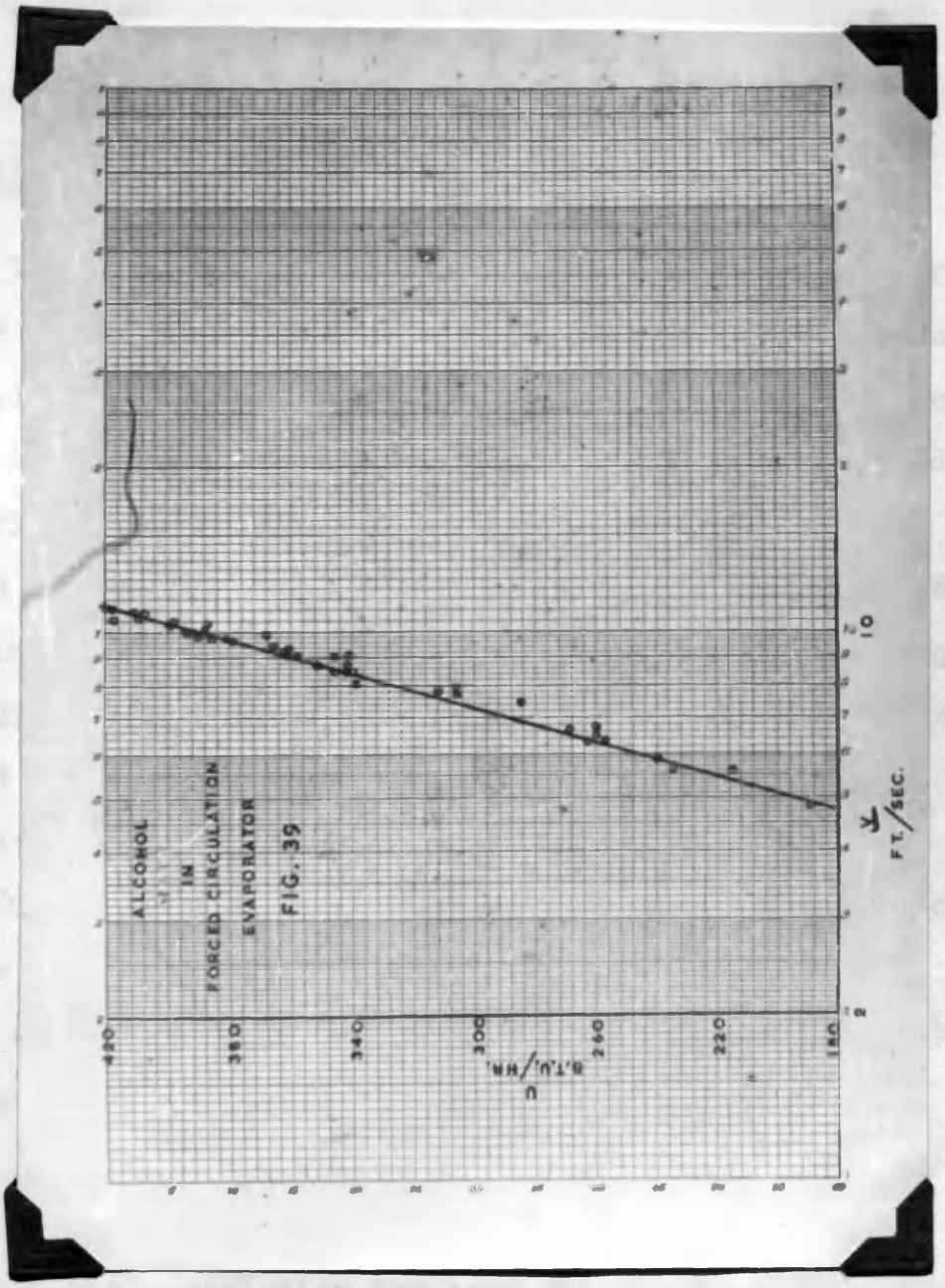

ALCOHOL
IN
FORCED CIRCULATION
EVAPORATOR
FIG. 39
U
B.T.U./HR.
420
380
340
300
260
220
180
2
10
V
FT./SEC.

FIG. 39

PART II.

SECTION C.

(a) LIQUID FILM COEFFICIENT FOR TOLUENE:

The apparatus was dismantled and dried. The orifice meter was then calibrated for toluene in the same way as described under alcohol. The calibration curve is shown in Fig. (37). The travelling thermocouple was calibrated with the fixed thermocouples at the inlet and outlet of the heating water at 80° C. Some 45 runs were made with feed rates ranging from 741 lbs./hr. to 1992 lbs./hr. representing the velocity range from 3.9 ft./sec. to 10.5 ft./sec. A vacuum of 58 cms. of mercury was maintained in all the runs. The latent heat of vapourisation of toluene was calculated from the Clausius-Clapeyron equation in the same way as described under toluene in natural circulation evaporator. All the rest of the calculations were done in the same way as before. The data are shown in Table (31).

(b) CORRELATION:

In the correlation the same principle was followed as in the two previous cases. All the values of the physical properties/

TABLE - 31.

TOLUENE IN FORCED CIRCULATION COPPER TUBE EVAPORATOR.

Heating Water at 80° C.

Vacuum - 58 cms. of mercury.

No Boiling in any Test.

Test No.	Feed Rate litres /hr.	Feed Rate lbs. /hr.	Evapn. lbs. /hr.	% Evapn.	Temp. in Separator. °F	Q B.Th.U./hr.	ΔT_m °F	U B.Th.U./hr.	h B.Th.U./hr.
1.	402	741	12.92	1.744	157.1	2145	13.14	128.0	135.0
2.	420	774	13.62	1.760	155.8	2260	13.42	132.1	139.5
3.	451	831	14.60	1.758	157.8	2424	12.60	150.8	160.5
4.	452	833	14.83	1.780	157.8	2461	12.53	154.1	164.2
5.	465	857	12.54	1.464	159.4	2082	11.61	140.6	149.0
6.	466	859	14.68	1.710	158.0	2435	12.48	153.1	163.0
7.	540	995	17.86	1.796	157.8	2965	12.51	186.0	201.0
8.	600	1105	20.60	1.866	155.8	3420	11.62	230.7	254.1
9.	607	1118	21.30	1.906	158.2	3533	12.44	222.8	244.6
10.	624	1150	21.38	1.858	158.5	3545	12.22	227.6	250.4
11.	654	1206	21.90	1.816	158.5	3635	12.40	230.0	253.3
12.	655	1207	22.13	1.834	159.0	3705	12.20	238.3	263.4
13.	713	1314	26.11	1.988	157.6	4335	12.76	266.5	298.3

Continued/-

TABLE - 31 (Continued)

Test No.	Feed Rate litres /hr.	Feed Rate lbs. /hr.	Evapn. lbs. /hr.	% Evapn.	Temp. in Separator °F	Q B.Th.U./hr.	ΔT_m °F	U B.Th.U./hr.	h B.Th.U./hr.
14.	739	1362	28.31	2.080	157.8	4700	12.66	291.0	329.3
15.	745	1373	26.60	1.938	156.7	4412	13.34	259.2	289.2
16.	768	1415	23.95	1.692	161.2	3974	11.04	282.5	318.5
17.	769	1417	28.78	2.030	157.9	4780	12.84	291.8	330.3
18.	780	1437	28.30	1.970	156.0	4698	13.76	268.0	302.0
19.	808	1489	29.75	1.998	157.8	4935	13.25	292.0	330.6
20.	820	1512	29.63	1.960	158.4	4916	13.15	293.3	332.3
21.	842	1552	31.01	1.998	156.0	5144	13.72	294.3	333.6
22.	847	1561	30.50	1.956	158.5	5060	12.21	325.3	374.0
23.	858	1581	30.32	1.918	159.1	5030	12.46	316.5	362.4
24.	864	1591	28.90	1.816	159.8	4798	11.93	315.1	360.6
25.	865	1593	35.15	2.206	156.2	5835	13.60	336.5	388.9
26.	880	1621	28.50	1.758	161.3	4732	10.92	340.2	393.8
27.	917	1690	31.90	1.889	158.2	5296	12.78	325.0	373.6
28.	927	1708	37.42	2.192	158.9	6210	12.67	384.5	454.4
29.	937	1726	30.78	1.784	158.4	5110	11.84	338.5	391.5
30.	958	1766	34.10	1.932	159.4	5668	12.13	366.0	428.8

Continued /-

TABLE - 31 (Continued)

Test No.	Feed Rate litres /hr.	Feed Rate lbs. /hr.	Evapn. lbs. /hr.	% Evapn.	Temp. in Separa-tor. °F	Q B.Th.U./hr.	ΔT_m °F	U B.Th.U./hr.	h B.Th.U./hr.
31.	964	1776	39.41	2.219	156.6	6540	13.44	382.0	450.9
32.	973	1792	39.35	2.196	159.1	6530	12.38	414.0	496.1
33.	977	1800	33.76	1.876	160.9	5601	11.49	382.5	451.6
34.	982	1809	29.75	1.644	161.2	4938	11.28	343.0	397.6
35.	995	1833	38.05	2.075	157.5	6310	12.97	382.0	451.0
36.	1022	1884	31.65	1.681	160.7	5250	11.40	361.0	422.0
37.	1025	1889	34.37	1.820	158.5	5700	12.30	364.0	426.0
38.	1026	1891	37.23	1.969	160.5	6180	12.08	401.0	477.6
39.	1035	1908	41.90	2.197	155.7	6950	13.44	406.0	484.7
40.	1040	1917	47.40	2.472	154.4	7861	15.18	406.0	484.7
41.	1057	1947	33.26	1.710	160.9	5520	11.33	382.0	451.0
42.	1066	1964	34.30	1.748	158.4	5690	12.11	368.5	431.8
43.	1072	1975	34.40	1.743	161.1	5710	11.55	387.8	459.0
44.	1080	1990	58.14	2.922	155.3	9654	14.98	505.0	632.8
45.	1081	1992	43.50	2.185	159.0	7220	12.03	471.0	580.3

properties of toluene were either taken from the appropriate tables or calculated from the proper equation from the International Critical Tables. The relevant calculated values are shown in Table (32). A plot of

$$\frac{hD/K}{(\mu C/K)0.4} \quad \text{against} \quad (DV\rho/\mu)$$

shows (Fig.40) a very high value of the power function of the Reynolds number, 1.25. The equation which satisfies the observed film coefficients is

$$hD/K = 0.000078\ (DV\rho/\mu)^{1.25}\ (\mu C/K)^{0.4}$$

In the calculation of the film coefficients, latent heat of vapourisation of toluene was calculated from the Clausius-Clapeyron equation and it was first thought that the higher power function of Reynolds number was due to, possibly, higher values of film coefficients thus calculated. But it will be remembered that in the natural circulation evaporator, the boiling film coefficients of both alcohol and toluene could be represented by a single equation. So it is unlikely that the above arguments are true. A plot of log V against the overall heat transfer coefficient, U, shows (Fig.41) the same nature as in the two previous cases.

TABLE - 32

TOLUENE IN FORCED CIRCULATION EVAPORATOR.

Test No.	Vel. ft/sec. V	$\frac{DV\rho}{\mu}$	$\frac{hD}{K}$ Obs.	$\frac{\frac{hD}{K}}{\left(\frac{\mu C}{K}\right)^{0.4}}$	a Cal.
1.	3.92	34030	62.7	34.35	0.0000751
2.	4.10	35600	64.8	35.50	0.0000728
3.	4.40	38200	74.6	40.85	0.0000770
4.	4.41	38290	76.3	41.80	0.0000784
5.	4.53	39350	69.2	37.90	0.0000687
6.	4.54	39430	75.7	41.45	0.0000746
7.	5.26	45670	93.4	51.15	0.0000772
8.	5.85	50800	118.0	64.60	0.0000862
9.	5.92	51390	113.6	62.20	0.0000805
10.	6.08	52780	116.3	63.68	0.0000802
11.	6.38	55400	117.8	64.50	0.0000760
12.	6.39	55500	122.4	67.00	0.0000788
13.	6.95	60330	138.7	75.90	0.0000803
14.	7.21	62600	153.2	83.80	0.0000851
15.	7.26	63030	134.4	73.50	0.0000739
16.	7.48	64930	148.0	81.00	0.0000785
17.	7.50	65100	153.6	84.00	0.0000810
18.	7.60	65980	140.4	76.80	0.0000731
19.	7.88	68430	153.6	84.10	0.0000765
20.	7.99	69400	154.4	84.50	0.0000753
21.	8.21	71300	155.1	84.85	0.0000734

Continued /-

TABLE - 32 (Continued)

Test No.	Vel. Ft./sec. V	$\frac{DV\rho}{\mu}$	$\frac{hD}{K}$ Obs.	$\frac{\frac{hD}{K}}{\left(\frac{\mu C}{K}\right)^{0.4}}$	a Cal.
22.	8.26	71700	173.8	95.00	0.0000808
23.	8.36	72600	168.4	92.30	0.0000786
24.	8.42	73100	167.6	91.70	0.0000765
25.	8.43	73200	180.7	98.90	0.0000823
26.	8.58	74500	183.0	100.20	0.0000815
27.	8.94	77660	173.8	95.00	0.0000738
28.	9.04	78500	211.2	115.60	0.0000890
29.	9.14	79380	181.9	99.70	0.0000756
30.	9.34	81100	199.0	109.00	0.0000800
31.	9.40	81600	209.5	114.80	0.0000832
32.	9.48	82300	230.5	126.20	0.0000906
33.	9.53	82730	209.8	115.00	0.0000824
34.	9.57	83100	184.8	101.30	0.0000718
35.	9.70	84200	209.6	114.80	0.0000803
36.	9.97	86600	196.2	107.50	0.0000727
37.	10.00	86870	198.0	108.50	0.0000728
38.	10.00	86870	222.0	121.60	0.0000812
39.	10.10	87700	225.2	123.40	0.0000815
40.	10.15	88100	225.2	123.40	0.0000812
41.	10.31	89500	209.6	114.80	0.0000742
42.	10.39	90200	200.5	109.90	0.0000705
43.	10.45	90700	213.4	116.80	0.0000745
44.	10.53	91400	294.0	161.00	0.0001006
45.	10.54	91500	267.6	146.60	0.0000923
					Av. -0.000078

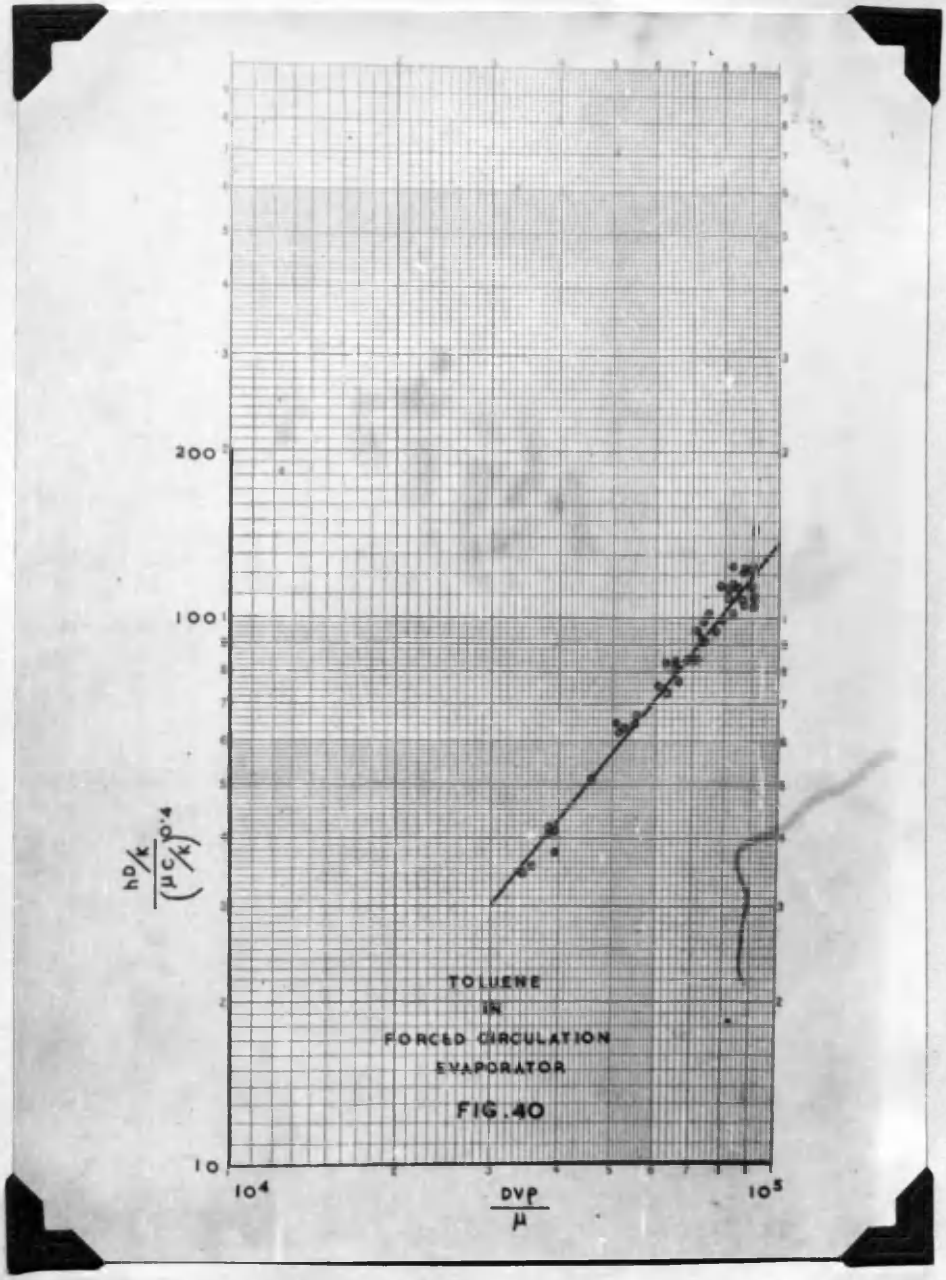

200
100
10
$\frac{hD/k}{(\mu c/k)^{0.4}}$
10^4
10^5
$\frac{DVP}{\mu}$
TOLUENE
IN
FORCED CIRCULATION
EVAPORATOR
FIG. 40

FIG. 40

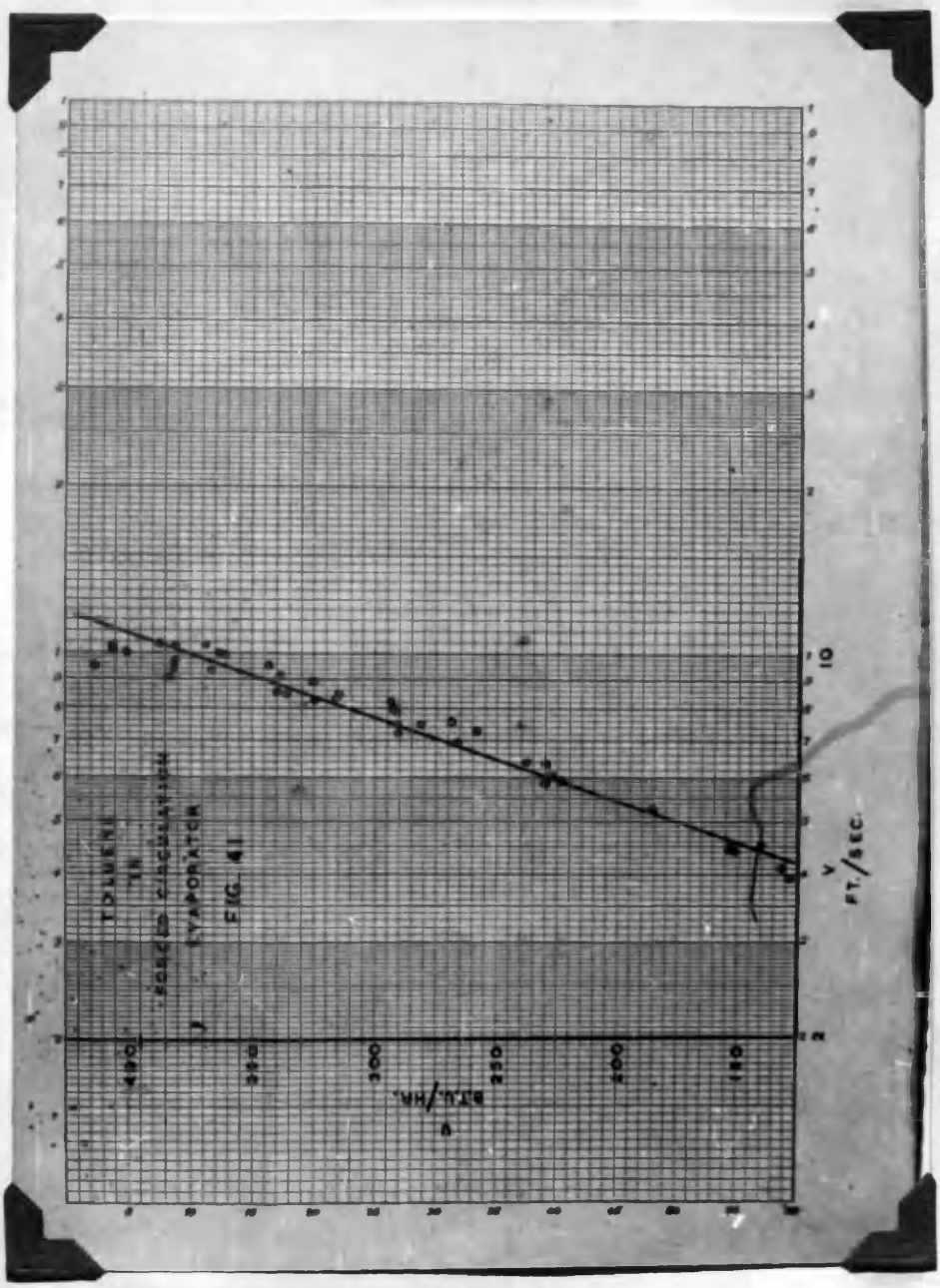

TOLUENE IN FORCED CIRCULATION EVAPORATOR
FIG. 41
B.T.U./HR.
V FT./SEC.
400
350
300
250
200
150
2
10

FIG. 41

PART II.

SECTION D.

DISCUSSION ON FORCED CIRCULATION EVAPORATOR.

From a study of the Tables (27,29,31) and also from Figs. (35,39 & 41) it will be seen that the overall and the film coefficients of all the three liquids increase with the increase of rates of feed. There is a remarkable constancy in the percentage evaporation in any set of experiments irrespective of the rate of circulation. The approximate percentage evaporation in water, alcohol and toluene are 1, 0.9 and 1.8 respectively. From a comparison of the results for each liquid both in the natural circulation and forced circulation evaporators it will be seen that the film coefficients in the forced circulation at higher rates of velocity reach the values of boiling film coefficients observed in the natural circulation. Of course, the natural circulation coefficients were at a limited range of ΔT which might have varied more widely. At very high rates of circulation in the forced circulation evaporator, the film coefficients obtained for water are as high as the boiling film coefficients and in few cases are higher than the average boiling film coefficients. Of course, these values, which are higher than the boiling film coefficients, are questionable/

questionable. In the case of toluene the film coefficients increased with the increased rates of circulation and approached the values of the boiling film coefficients but never exceeded them. Most noticeable features were observed in the values of alcohol. The film coefficients increased with the increased rate of circulation and the values were higher than the boiling film coefficients. The overhead pressures and ΔT in both the natural and forced circulations were not very different as to explain such results. Other interesting points in this respect are that whereas the boiling film coefficients of alcohol and toluene in the natural circulation copper tube evaporator could be represented by a single equation, the film coefficients of water and alcohol in the forced circulation evaporator could be represented by a single equation. Thus alcohol being the common factor in both the evaporators, the coefficients of which could be represented along with those of one of the other two liquids by a single equation, it becomes very difficult to find out the reasons which are responsible for giving values of film coefficient in the forced circulation evaporator higher than the boiling film coefficients in the natural circulation evaporator.

Usually the flow conditions, in which the main stream flows in turbulent motion, resemble those shown in the diagram

(42)/

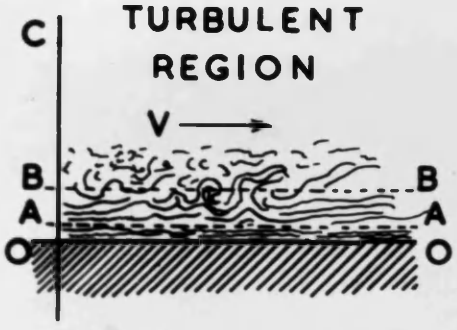


FIG. 42

(42). At the wall OO the velocity is zero; and very near the wall, OA, the fluid is moving in laminar motion. A colour band introduced at the axis is quickly dissipated by the turbulence but a second colour band, laid down simultaneously near the wall, is not disturbed was shown by Couch and Herrstrom (18). In this buffer layer AB, the motion may be either streamline or turbulent at a given instant. In the zone BC, the motion is always turbulent. The thickness of the buffer layer, AB, varies with time because of the more or less periodic formation of vortexes. Though the thickness of the film OA is small, increasing the velocity of the main stream makes a substantial and important reduction in its thickness.

In the film flowing along the wall the molecules of the fluid, which make it, are by no means stagnant, i.e. under circumstances they can move to and from the film giving place to new ones. In streamline flow, the chance of this replacement is very less. So most of the heat leaving the wall, in streamline flow, is conducted through the film to the buffer layer. A fraction of the heat is consumed in warming the film and this portion does not reach the buffer layer. In the buffer layer, part of the heat is transferred by convection, i.e. by mechanical mixing due to the eddies, the remainder of the heat is transferred by conduction under the influence of radial gradient/

gradient in temperature. Though the cross section of the film is negligible compared to that of the entire tube and the fraction of heat carried away by the film and the buffer layer is negligible, the film plays an important role on the entire mechanism of heat transfer. In stream-line flow, the particles do not freely move from and to the film, and as the conduction of heat through liquid films is very low, the film gives much resistance to heat flow from the wall to the buffer layer. If the heated particles in the liquid film could be allowed to move frequently giving place to new particles, the resistance to heat flow from the wall through the film would be less as in that case more heating would be by actual convection. In turbulent motion this is possible. The film thickness is decreased and the buffer layer gives considerable eddy formation which in turn agitates the molecules in the film and gives an opportunity of free mixing of the heated particles from the film. This is the reason of increased rate of heat transfer in forced circulation.

The liquid enters the evaporator at substantially the equilibrium temperature in the vapour space, although in some cases it may be somewhat cooler than this temperature due to heat losses in the return line. Heat is transferred through the tube wall and superheats the liquid above the equilibrium temperature/

temperature in the vapour space. The liquid is kept from vapourising by the hydrostatic and friction heads above it. In this region the Dittus-Boelter equation holds and the heat pssses by conduction through the viscous film and by convection through the bulk of the liquid. It is known that the dropwise condensation of steam can produce coefficients 4 to 8 times that of the viscous condensation. If the production of a bubble of water from steam can be so much more effective than transferring heat through a viscous film of condensate on the outside of the tube, then it seems possible that the production of a bubble of steam in the inside of the tube would likewise be a better heat transmission agency than a viscous film. More of these steam bubbles are formed inside the tube the better will be the rate of heat transfer. The minute bubbles of steam so formed are removed immediately and convection into the cooler bulk of the liquid condenses them. As the superheated liquid passes up the tube it becomes still hotter. This has the effect of raising both the Prandtl number and the Reynolds number. Consequently, the heat transferred through the viscous film increases because of scouring effect. If the velocity of the liquid is not high, boiling occurs at a point where the hydrostatic and friction heads are not sufficient to prevent vapourisation of the liquid. An evolution of bubbles occurs/

occurs which tremendously increases the volume of the mixed liquid and vapour. This induces a large velocity which reduces the thermal resistance of the liquid film over the part of the tube in boiling action. If the velocity of the liquid is high, no boiling occurs (as in the present work) and the vapourisation takes place by flashing in the vapour head of the evaporator. This latter process demonstrates the predominant mechanism of the forced circulation evaporator to be simple heating.

So far as the rate of heat transfer is concerned, the forced circulation may be of great advantage. But from an economic standpoint, it is necessary to balance the cost of rapid circulation against the more rapid evaporation. "Fifteen feet per second for most things is a velocity so high that the cost of power is greater than the decreased size of the evaporator due to increased coefficient" - (Badger in discussion of paper of Boarts et al). But conditions differ at each location. A location which has cheap power could undoubtedly afford to go to a higher rate of circulation than another location where power costs are high. Considering all the factors, not solely the heat transfer, that are involved in the choice of commercial machines, for many purposes a long/

long tube natural circulation vertical evaporator is better. However, the forced circulation evaporator has one special field, handling of liquids which tend to deposit scale or crystals on the heating surfaces, foamy liquids, viscous liquids, etc., which cannot be handled in the natural circulation type evaporator. That sufficiently limits the application of the forced circulation evaporator.

P A R T III

LATENT HEAT OF VAPOURISATION OF TOLUENE.

PART III

DETERMINATION OF LATENT HEAT OF VAPOURISATION OF TOLUENE AT VARIOUS TEMPERATURES.

Actual determined values of the latent heat of vapourisation of toluene at various temperatures are not available in the literature. In the calculation of heat transfer in the case of toluene the values of the latent heat of vapourisation, at the temperatures at which the experiments were carried out, were obtained from a graph based on the Clausius-Clapeyron equation. It was, therefore, thought desirable to determine the latent heat of vapourisation of toluene at the temperatures at which the experiments were carried out. An apparatus was designed, with electrical equipment, which could be used for this purpose. A sketch of the apparatus is shown in Fig. 43.

APPARATUS.

A is a 250 c.c. pyrex round bottom flask with a male ground glass joint B in the middle and protruding neck C above the ground glass joint. There are two side tubes D D joined in the fashion as shown, in order to hold a pool of mercury in each arm. E is a glass tube about 4 inches long and $\frac{5}{8}$ inch in diameter and with a narrow mouth. At the bottom /-

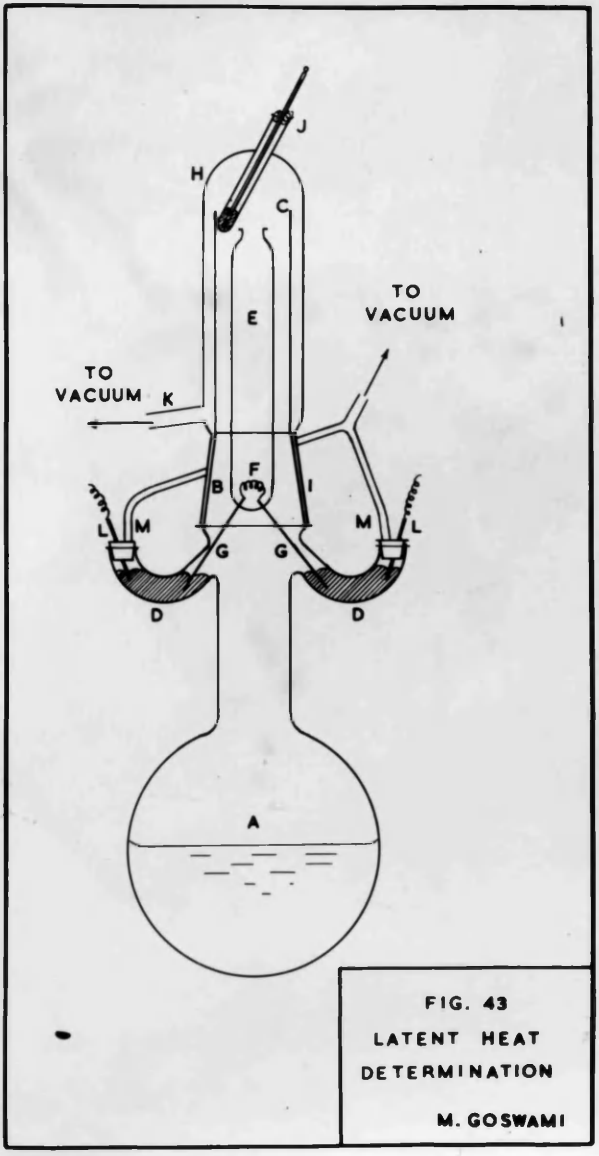

J
H
C
E
TO
VACUUM
TO
VACUUM
K
F
B
I
L
M
M
L
G
G
D
D
A
FIG. 43
LATENT HEAT
DETERMINATION
M. GOSWAMI

FIG. 43

bottom of this tube is a coil of platinum wire, F, about 10 inches long and about 0.1 mm in diameter. This platinum coil is fused to two other thick wires of platinum G G fused through the bottom of the glass tube E. These two platinum leads G G are dipped into the pools of mercury D D allowing electrical connection. H is a covering glass tube with a female joint of ground glass, I, a thermo-pocket J to register the temperature and a side tube K leading to the vacuum line. L L are two thick iron wires connected with the mercury in D D. M M are two small tubes leading to the vacuum line in order to evacuate the air left in those two arms, thus equalising the pressure in the flask and in these two arms. There is an ammeter graduated in 0.02 amp. from 0 to 1 amp. and a voltmeter graduated in 0.5 volt from 0 to 20 volts.

WORKING OF THE APPARATUS.

Half of the flask A is filled up with the liquid whose latent heat of vapourisation is to be determined together with some glass beads to prevent bumping of the liquid. A suitable and known amount of the same liquid is taken in the tube E and the whole apparatus is arranged as shown in Fig. 43. An ordinary condenser is fitted at K.

The /-

The liquid in A is allowed to boil for about 20 minutes by which time the liquid in E is brought to the temperature of the boiling point of the liquid by the hot vapour rising from the flask A. It is then allowed to cool and the whole tube E is brought out and weighed. The difference between the first and the second weight gives the amount of liquid lost by evaporation due to heating. The tube E is replaced and again the liquid in A is allowed to boil for the same period of time as in the first case. Then a current is passed through F for a known period of time noting the current and voltage every minute. The success of the experiment depends on the immediate starting of boiling as soon as the current is switched on and vice-versa. Generally the current is passed for about 7 minutes. It is then allowed to cool and the tube E is weighed again. The difference between the second weight and the third weight minus the loss due to initial heating is the actual amount of liquid evaporated by the current used. The latent heat of vapourisation is calculated thus:

$$\text{Latent heat} = \frac{\text{Total heat reqd. for evaporation at the b.pt.}}{\text{Amount of liquid evaporated.}}$$

$$= \frac{\text{Volt} \times \text{Amp.} \times \text{Seconds}}{4.183 \times \text{loss due to current}} \quad \text{gm. cal./gm.}$$

To make sure of the reliability of the apparatus, two determinations were first made on benzene whose latent heat of vapourisation is known (94.17 gm.cal./gm. at 80°C., the boiling point of benzene at atmospheric pressure). The values obtained are given in Table 33. The values obtained are very satisfactory.

After this, experiments were carried out with toluene at the atmospheric pressure and the values of latent heat of vapourisation obtained were near about the values mentioned in the books. So being satisfied with the performance of the apparatus at atmospheric pressure, attempts were made to determine the latent heat of vapourisation at temperatures below that of the boiling point at atmospheric pressure.

DETERMINATION OF LATENT AT LOW PRESSURES.

The side tube K of the apparatus was connected, through an ordinary condenser and a mercury vacuum gauge with a release valve to regulate the pressure, to the vacuum pump. Under reduced pressure also the same method of carrying out the experiments was followed.

Many /-

TABLE - 33

LATENT HEAT OF VAPOURISATION OF BENZENE AT ATMOSPHERIC PRESSURE.

Test No.	Initial loss of wt. gm.	Final loss of wt. gm.	Actual loss of wt. gm.	Volts.	Amp.	Time mins.	Latent Heat cal/ gm.
1.	0.0600	2.6916	2.6316	4.0	0.865	5	94.40
2.	0.0567	4.2362	4.1795	4.0	0.859	8	94.15
In literature at boiling point at atmospheric pressure						-	94.17

Many difficulties were encountered during the experiments under reduced pressure. Various methods were tried to prevent the liquid from bumping; glass beads, glass wool, porous porcelain beads and powder, very small and fine capillary tubes etc. were tried. Whereas the glass beads prevented the bumping of the liquid in the flask A, nothing was found suitable for the prevention of bumping of the liquid in E. Sometimes the boiling of the liquid in E started quite smoothly on switching on the current but stopped boiling after a while and much superheating of the liquid took place and after some time the liquid bumped so vigorously that even the packings or the boiling aid, whatever it might be, in E were thrown out of the tube. No suitable means of stopping this bumping of liquid in the tube E was found out at first.

An investigation into the reasons for this bumping of liquid might give some light into the mechanism of boiling. It has been observed by many investigators, as well as in the glass tube evaporator of the present work, the important role of dissolved air in boiling. At low pressure all the dissolved air is removed and as air helps in forming the centre of bubble formation, there is less chance of boiling to start. Also it is known that at low pressures

pressures /-

the rate of heat transfer is low and hence the liquid gains superheat gradually and when the liquid is sufficiently superheated, any sort of disturbance, even to the minutest degree upsets the whole system. Under the superheated condition when a small bubble of vapour starts forming, the superheat of the liquid gives all the energy needed to increase the size of the vapour bubble. The process is so rapid that a sudden formation of a large bubble of vapour upsets the system with explosive violence. This may be the possible reason of bumping of liquid at low pressures.

It was then proposed to find out the latent heat of vapourisation of toluene at different pressures of above the bumping level and to plot a graph of latent heat vs. pressure and to extrapolate the value of latent heat of vapourisation at the required pressure. Experiments were carried out according to this plan but the results obtained were not very encouraging as will be evident from the Table 34.

A deposit of carbon on the heating filament of platinum was tried, hoping that this sort of deposit might help the formation of vapor bubbles which in turn would give /-

TABLE - 34

LATENT HEAT OF VAPOURISATION OF TOLUENE.

Test No.	Loss in wt. gm.	Volts.	Amp.	Time mins.	Pressure mm/Hg.	Latent Heat cals./gm.
1.	4.2692	4.50	0.867	7	Atm.	90.80
2.	4.6649	4.45	0.867	7	"	83.10
3.	4.1868	4.40	0.854	7	"	90.01
4.	4.4056	4.33	0.855	7	"	82.50
5.	4.3390	4.50	0.865	7	"	90.10
6.	4.2516	4.38	0.867	7	"	89.70
7.	4.2142	4.35	0.867	7	"	89.80
8.	4.1401	4.20	0.855	7	496	87.20
9.	4.0368	4.20	0.854	7	496	89.20
10.	1.9030	4.72	0.865	3	402	92.40
11.	2.0700	5.20	0.870	3	402	94.10
12.	2.0200	4.98	0.875	3	402	92.80
13.	1.9700	5.00	0.863	3	301	94.40
14.	2.0816	5.25	0.870	3	301	97.70
15.	0.8680	3.25	0.594	3	282	95.70
At atmospheric pr. (109.6° C. b.p.) in literature -						86.53

give smooth boiling. A small quantity of the experimental liquid was taken in the tube E so as just to keep the element submerged. The apparatus was then assembled as usual and the liquid in the flask A was boiled till some of it distilled over. The current was then switched on and maintained till most of the liquid in the experimental tube E distilled away and the platinum filament began to glow. When cooled, a black deposit of carbon was found on the filament, possibly due to the cracking of the vapour by the hot glowing platinum filament. It was found that while this deposit of carbon gave a fairly smooth boiling at atmospheric pressure, it did not help much at low pressures. Finally, a deposit of very finely divided platinum on the heating filament was tried. The filament was first wetted with a strong solution of platinum chloride and then a current was passed through the filament so as to make it glow. When cooled, a very fine grey deposit was observed on the filament. Very small bubbles regularly evolved in chains from the surface of the filament and this deposit was found to give a very smooth boiling even at low pressures. To check its performance, the latent heat of vapourisation of benzene was again determined at atmospheric pressure. From a comparison of the results in Table 35, it will be seen that correct values were obtained with the platinum deposit.

TABLE - 35

LATENT HEAT OF VAPOURISATION OF BENZENE.

Test No.	Boiling Aid.	Actual loss in wt. gms.	Volts.	Amp.	Time mins.	Pr. mm/ Hg.	Latent Heat cal/gm.
1.	Glass beads.	2.162	2.80	0.87	6	Atm.	97.0
2.	"	2.326	2.90	0.89	6	"	95.5
3.	Pt.	1.864	2.82	0.87	5	"	94.4
4.	"	1.901	2.84	0.88	5	"	94.3

Experiments were then carried out for the determination of latent heat of vapourisation of toluene, at atmospheric and low pressures, with this platinum deposited filament. The results obtained with various boiling aids and at different pressures are shown in table 36. It will be seen that the three tests made on toluene at atmospheric pressure with this platinum deposited filament (viz. Nos. 9. 10. & 11) give an average value of 86 gm. cal./gm. (actual value at 110.6° C. is 86. 5 gm. cal./gm.) while the values obtained with other aids are very divergent at all pressures including atmospheric. Even with platinum deposit as the boiling aid consistent results could not be obtained with this apparatus at low pressures. It is most probable that conditions at low pressures are so unstable that it is very difficult to obtain a steady vapourisation. At pressures below 100 mm. of mercury the boiling was found to be so sensitive to pressure that even a very slight change of it altered the rate of boiling. It would seem, therefore, that unless the pressure in the system could be held absolutely steady, no consistent values could be obtained at low pressures. Unfortunately , lack of time did not permit further continuation of the work in this line.

TABLE - 36

LATENT HEAT OF VAPOURISATION OF TOLUENE.

Test No.	Boiling Aid.	Actual loss in wt. gms.	Volts.	Amp.	Time in mins.	Pr. in mm/ Hg.	Latent Heat cal/gm.
1.	Glass beads.	2.939	5.40	0.866	4	Atm	91.3
2.	"	1.871	3.03	0.891	4	"	82.8
3.	"	1.288	2.97	0.889	3	"	88.2
4.	Carbon	1.770	3.10	0.891	4	"	89.6
5.	"	1.837	3.01	0.888	4	"	83.2
6.	Pt.	1.803	2.50	0.881	5	"	87.6
7.	"	2.297	2.50	0.883	6	"	82.7
8.	"	1.400	2.60	0.888	3½	"	82.8
9.	"	2.290	2.92	0.938	5	"	85.8
10.	"	2.270	2.91	0.932	5	"	85.6
11.	"	2.222	2.90	0.923	5	"	86.5
12.	Glass beads	1.666	3.05	0.871	4	458	91.5
13.	"	0.868	3.25	0.594	4	282	95.7
14.	Pt.	1.325	2.35	0.868	4	98	90.2
15.	"	1.011	2.32	0.880	3	98	86.9
16.	"	1.311	2.37	0.892	4	96	92.6
17.	"	1.293	2.38	0.893	4	90	94.3
18.	"	1.242	2.35	0.885	4	90	96.0

SUMMARY & DISCUSSION.

SUMMARY & DISCUSSION

For the determination of boiling film coefficients of heat transfer, conditions were arranged so that all the thermal resistances except that of the film in question remained constant. Hot water at a constant temperature of 50° C. and 80° C. and a constant high velocity of about 10 ft. per second was circulated through the outer jacket and water at different known velocities was passed through the inner tube comprising the evaporator. The inner liquid film coefficient can be determined from the Dittus-Boelter equation

$$h = a \frac{K}{D} \left(\frac{DV\rho}{\mu}\right)^{0.8} \left(\frac{\mu C}{K}\right)^{0.4}$$

Excepting the viscosity, the properties of the liquid comprising the film vary only slightly with temperature. Therefore, the equation can be written as

$$h = K' \frac{V^{0.8}}{\mu^{0.4}}$$

By plotting $1/U$ against $\mu^{0.4}/V^{0.8}$ a straight line was obtained which when extrapolated to cut the Y-axis represented the value of $1/U$ at infinite velocity of feed liquid /-

liquid or at a value of 1/h = 0. This value of the intercept represented the combined annular film and the wall coefficients as under those conditions

$$\frac{1}{U} = \frac{1}{H_A} + \frac{1}{H_W}$$

The value of the intercept for the combined annular film and wall coefficients obtained with the stainless steel tube are

At 50°C. heating water = 0.0015

80°C. .. .. = 0.0013

which suggest value for the combined coefficients to be 667 B.Th.U./hr. and 770 B.Th.U./hr. respectively. These low values are due to the comparatively low conductivity of steel. Values of the same order have been obtained in the Kestner evaporator, the material of the tube being of copper and the experiments carried out at 5 lbs. and 10 lbs. of steam pressure per sq. in. This low value in the case of the Kestner evaporator is attributed to the accumulation of scale on the outside of the evaporator tubes. In the copper tube, of course, the values obtained were more favourable. The values of the intercepts obtained are

At 50°C. heating water = 0.000362

80°C. .. .. = 0.000338

Taking /-

Taking the conductivity of copper to be 220 B.Th.U./(hr.) (sq.ft.) (^{o}F./ft.), the annular film coefficients, i.e. the film coefficients of the heating water, calculated are

At 50^{o} C. = 2950 B.Th.U./(hr.) (sq.ft.)(^{o}F.)

80^{o} C. = 3200 .. ..

These are reasonably high values. The copper tube was used to centre the whole resistance to heat transfer in the film investigated. After using the copper tube for some time a second determination of the annular film and wall coefficients gave slightly lower values, suggesting a film deposit on the tube wall.

In determining the boiling film coefficients of the liquids, the overall boiling heat transfer coefficients were first determined. Then substituting the value of the intercept obtained from the graphs, the boiling film coefficients were calculated from the relation

$$\frac{1}{U_B} - \left(\frac{1}{H_A} + \frac{1}{H_W}\right) = \frac{1}{h_B}$$

A differential thermocouple arrangement was made with three thermocouples which registered difference of temperature between the heating water and the liquid at different levels in the evaporator tube. Two of the thermocouples /-

thermocouples were placed at the inlet and outlet of heating water while the third was used as a travelling thermocouple up and down the evaporator tube in a thermocouple pocket. The average temperature difference in the section concerned was obtained by graphical integration of a plot of temperature difference vs. distance from bottom of the evaporator tube.

Boiling film coefficients of water have been determined in natural circulation evaporator with stainless steel tube and copper tube under low pressure and in a Kestner evaporator at atmospheric pressure. The primary variable in all the sets of experiments was the rate of feed. Plots of U_B against pounds of evaporation per hour (Fig. 29) for all the three evaporators show a perceptible fall in U_B with the increase in evaporation in the steel and the copper tube evaporators but not so marked a fall in the Kestner evaporator. The possible reason for this might be that at higher evaporation part of the tube in the boiling section is blanketed by the vapour bubbles. In the Kestner evaporator this effect is not pronounced because of the wetting action of the falling film of liquid in the second section. The average value of U_B is round about 500 B.Th.U /hr. for steel tube and the Kestner evaporators, and 1200 B.Th.U./hr. for /-

for the copper tube evaporator. U_{NB} in the copper tube evaporator is about 550 B.Th.U./hr. Without giving too much weight to the extreme values obtained, the values of h_B, the boiling film coefficient, in the steel tube is about 1800 B.Th.U./hr. and about 2000 to 3000 B.Th.U./hr. in the copper tube evaporator. In the Kestner evaporator the value of h_B has been more varied, possibly due to the uncertain nature of condensation of steam in the steam chest, and in a few cases reached unreal values. The following equation correlated the observed boiling film coefficients in the copper tube evaporator :

$$h_B = \frac{8.2 \times 10^8 \times v^{0.1}}{\left(\frac{\mu C}{K}\right)^{0.3} \times \sigma^2 \times (\Delta T_B)^2}$$

If the power factor of ΔT_B was substituted by 1.75, the equation correlated the results of h_B for water observed in the stainless steel tube evaporator. In the correlation though the term for feed rate per hour does not occur, all the terms involved are dependent on the independent variable feed rate. In other words this shows that the feed rate influences the boiling film coefficient. It is realised that this correlation of the observed boiling film coefficients could be made without bringing the Prandtl group and the surface tension into the equation as the properties of water /-

water do not change very much at the different average temperatures of the liquid in different runs. The observed boiling film coefficients in the Kestner evaporator could be correlated by an equation

$$h_B = \frac{1.692 \times 10^6 \times T_f}{A_B \; W_E \; (W_f)^{1.5}}$$

The correlation was not very satisfactory because of the varied values of the boiling film coefficients observed, possibly due to the uncertain nature of the coefficient on the steam side. This uncertain nature of the film coefficient of the heating medium was altogether eliminated in the steel tube and copper tube evaporators by circulating hot water at a constant high speed and a constant temperature. It is stressed here that these are purely empirical equations and can only be used for experiments done in the same type of apparatus and under similar conditions.

Boiling film coefficients of alcohol and toluene have been determined in the natural circulation copper tube evaporator at low pressure. The boiling film coefficients of toluene increase regularly with the increase of feed rate and decrease regularly with the increase of overall temperature drop in the boiling section (Figs. 19 & 20). The value of h_B for alcohol does not change up to a value of feed rate of about 40 lbs./hr.; after this value is reached the /-

the value of h_B increases with the increase of feed rate. On the other hand, the value of h_B decreases with the decrease of overall temperature drop in the boiling section only up to a value of about 300 B.Th.U./hr. After this value is reached, ΔT_B shows no effect on h_B. On an average a value of about 300 B.Th.U./hr. and 550 B.Th.U./hr. can be assigned to the boiling film coefficients of alcohol and toluene respectively under the conditions under which the experiments were carried out. Correlations of the boiling film coefficients of alcohol and toluene have been made with individual equations.

$$h_B = \frac{30.63 \ (W_f)^{0.25} (\Delta T_B)^{0.85}}{\left(\frac{\mu C}{K}\right)^{0.3}} \quad \text{for alcohol}$$

and

$$h_B = \frac{1696 \ (W_f)^{0.33}}{\Delta T_B} \quad \text{for toluene.}$$

76% of the data for alcohol and 92 % of the data for toluene fall within $\pm$ 10 % of the calculated values according to these equations. The influence of rate of feed per hour and ΔT_B on the boiling film coefficients are shown. While their effects are complimentary in the case of /-

of alcohol, they are contradictory in the case of toluene. A single equation

$$h_B = \frac{535 \quad (W_f)^{0.065}}{\left(\frac{\mu C}{K}\right)^{0.3} \quad A_B}$$

was also developed to correlate the boiling film coefficients for both the liquids. 75 % of the calculated values of h_B for alcohol and 90 % for toluene fall within ± 10 % of the observed values of h_B. This equation demonstrates the influence of boiling area and the feed rate per hour on the boiling film coefficients of these two liquids. It is to be mentioned here that in the calculation of the overall heat transfer coefficients for toluene, the latent heat of vapourisation at the temperature concerned was obtained from the Clapeyron-Clausius equation, as it was not elsewhere available. This correlation justifies the action taken.

To study the nature of boiling and the mechanism of of heat transfer, a pyrex glass evaporator was installed. Water, Perminal W.A. (a frothing agent) solution in water, alcohol, and toluene were studied in this evaporator. Three types of action comparable with those described by Brooks & Badger (13) were observed. Dissolved air in the feed liquid was /-

was found to agitate the liquid in the non-boiling section and perhaps increase the heat transfer coefficient in that zone. When the feed liquid was free from air, the nature of boiling was influenced. Irrespective of the rate of feed, the bubbles of vapour formed from an irregular surface in the tube and maximum temperature of the liquid was found to be much below the point where the formantion of vapour bubbles was visible. This phenomenon was very much pronounced in the case of alcohol and toluene. This suggests two things - firstly, the influence of air on the formation of vapour bubbles and secondly, that the maximum temperature of the liquid at a point does not always indicate that the boiling is taking place there. A distinctive change on the nature of boiling was observed when Perminal solution was used. There was no"slug" action, i.e. there were no alternate slugs of liquid and vapour in the longitudinal direction but instead there was an almost continuous column of bubbles in the whole of the boiling section of the evaporator tube and the line of demarcation between the boiling and the non-boiling sections was quite distinct (Fig. 16). Incidentally some determinations of the overall heat transfer coefficients in the glass evaporator were made. A value of about 102 B.Th.U./hr. for the /-

for the overall heat transfer coefficients based on the entire length of the tube was obtained for water. No improvement in the value of U was observed with solutions having lower surface tension. Of course the conditions in these two sets of experiments with water and Perminal solution were not identical. For U an average value of 55 B. Th. U./hr. for alcohol and 85 B. Th. U./hr. for toluene were obtained. There has been a remarkable constancy in these values obtained. Over a wide range of overall temperature difference, keeping the boiling length constant, overall heat transfer coefficients in the boiling section have been determined for water and alcohol. At constant boiling length there is a little increase in the value of U_B with the increase of ΔT_B (Fig. 14). The contrary observations were made by Cessna et al (15). For U_B on an average 110 B. Th. U./hr. and 78 B. Th. U./hr. were obtained for water and alcohol respectively.

In a forced circulation copper tube evaporator, film coefficients for water, alcohol and toluene have been determined. The rate of circulation was measured by an orifice meter, the velocities varying between 3.5 and 11 ft. per second. No boiling took place in any of the runs and the overall and the film coefficients of all the three liquids /-

liquids increased with the increased rate of circulation. The evaporation took place by the flashing of the liquids in the separator and the percentage of evaporation in any set of experiments was constant, being 1.0, 0.9 and 1.8 for water, alcohol and toluene respectively. This indicates the predominant mechanism of the forced circulation evaporator to be simple heating. At very high rate of circulation the film coefficients of water and toluene were as high as the boiling film coefficients observed in in natural circulation. In the case of alcohol, the film coefficients were higher than the boiling film coefficients in natural circulation. The reason is obscure, specially because of the facts that the film coefficients of water and alcohol in forced circulation could be represented by a Dittus-Boelter correlation

$$h = 0.0055 \frac{K}{D} \left(\frac{DV\rho}{\mu}\right)^{0.9} \left(\frac{\mu C}{K}\right)^{0.4}$$

and those of the boiling film coefficients of alcohol and toluene in natural circulation could be represented by the equation

$$h_B = \frac{535 \ (W_f)^{0.065}}{\left(\frac{\mu C}{K}\right)^{0.3} A_B}$$

The /-

The film coefficients of toluene in forced circulation could be correlated by a Dittus-Boelter relation

$$h = 0.000078 \frac{K}{D} \left(\frac{DV\rho}{\mu}\right)^{1.25} \left(\frac{\mu C}{K}\right)^{0.4}$$

The latent heat of vapourisation of toluene at different temperatures was determined in an electrical apparatus. The determination at atmospheric pressure was quite satisfactory but at low pressure it was not satisfactory at all. The nature of the surface of the heating element was important in giving a smooth boiling at low pressures.

From the observations on the glass tube evaporator and latent heat determination, the following inferences can be made. Since the dropwise condensation of steam gives coefficients 4 to 8 times that of the viscous film condensation, it can be expected conversely that production of bubbles of vapour inside the tube will increase the rate of heat transfer.

(1) Absence of air does not favour the formation of bubbles of vapour inside the tube.

(2) Absence of air allows the liquid to gain superheat to a great extent and when a bubble of vapour begins to form, the superheat of the liquid gives all the energy for /

for its growth. The process is so rapid that the sudden growth of a large bubble of vapour upsets the whole system altogether.

(3) Nature of heating surface is important for the formation of bubbles of vapour.

(4) In the natural circulation the liquid is heated to boiling which starts at a point where the hydrostatic and friction heads are not sufficient to prevent vapourisation of the liquid. In the non-boiling section the heat flows from the wall through a viscous film by conduction and then to the bulk of the liquid by convection. In the boiling section the superheat of the liquid is transformed to latent heat and vapourisation begins. The volume of the mixture of vapour and liquid increases and so also the velocity of flow in that section. This makes the film next to the wall thinner and the resistance of the film to heat flow decreases. Thus the rate of heat transfer in the boiling section is in general higher than that in the non-boiling section.

(5) As no boiling takes place in the forced circulation evaporator, the main mechanism is simple heating. The high /-

high velocity of circulation makes the film of liquid next to the wall thinner decreasing its resistance to heat flow. The value of the film heat transfer coefficient approaches those of the boiling film coefficients in the natural circulation.

CONCLUSIONS.

1. The division of a vertical tube evaporator into non-boiling and boiling sections is justified.
2. In general, three types of action in the evaporator tube comparable with those described by Brooks & Badger (13) have been observed.
3. The nature of the heating surface is important for the formation of vapour bubbles, especially at low pressures.
4. Dissolved air in the feed liquid plays an important role in boiling. Absence of air does not favour the formation of vapour bubbles. Results with air free liquid show that boiling does not always start at the maximum liquid temperature point.
5. A distinctive change on the nature of boiling is observed with solution having lowered surface tension. There are no alternate slugs of liquid and vapour but instead an almost continuous column of vapour bubbles in the whole of the boiling section of the tube and the line of demarcation between the boiling and the non-boiling sections is sharp.

6. The use of the combined annular film and wall coefficients for the determination of the boiling film heat transfer coefficient is quite satisfactory.

7. The overall temperature drop in the boiling section, the rate of feed and the boiling area were found to influence the boiling film heat transfer coefficient of the liquid.

8. In particular, at constant boiling length the overall heat transfer coefficient in the boiling section increases with the increase of overall temperature drop in the boiling section.

9. In forced circulation evaporation under non-boiling conditions at very high rates of circulation the film heat transfer coefficients of water and toluene reach values as high as their boiling film coefficients in natural circulation. The film coefficients of ethyl alcohol reach higher values than the boiling film coefficients in the natural circulation. All the forced circulation film coefficients can be represented by Dittus-Boelter equation with a higher value of the power function of the Reynolds number and a lower value of the constant "a".

10. There is a fall in the value of the overall heat transfer coefficient in the boiling section with increased rate of evaporation, due to a blanketing effect of the boiling section by the vapour bubbles. This blanketing effect has not been very pronounced in a special Kestner evaporator due to wetting action by a falling film of the liquid.

11. The temperature measurement of the liquid in the evaporator tube at different levels by a travelling thermocouple in a pocket is not a reliable guide to actual film temperature.

12. Circulation of hot water at a constant temperature and a constant high velocity gives a steady value of heat transfer from medium to tube wall. It is much more reliable than condensing steam.

SUGGESTION FOR FURTHER WORK

1. In the present work the influence of feed rate and boiling area were observed on the boiling film heat transfer coefficients of water, ethyl alcohol and toluene. It is very difficult to reduce the number of variables to manageable proportions and it is possible that work on one liquid would have been more revealing.
2. In the present work most experiments were carried out in the moderate heat flux region. Work with high heat flux should be carried, possibly by heating the tube electrically, the temperature of the tube surface next to the film being determined from the conductivity of the material of the wall.
3. A possible reason of bumping of liquid at low pressures has been given. A systematic and detailed study of the problem can possibly give some valuable information about the mechanism of boiling.
4. Dissolved air in the feed liquid was found to disturb the natural circulation in the non-boiling section and in the boiling section of the tube this air might possibly have a blanketing effect thus /-

thus affecting the rate of heat transfer. Industrial water is saturated with dissolved air and its influence on the rate of heat transfer should be fully investigated.

5. Recording of the temperatures of the liquid at different levels in the evaporator tube by a travelling thermocouple in a thermo-pocket is not a reliable guide to actual film temperature. Arrangements should be made so that the bare end of the travelling thermocouple comes in direct contact with the fluid inside the tube. Arrangements should also be made to explore the temperature distribution across the tube.

NOMENCLATURE

A - Heating area of evaporator tube considering the arithmetic mean of the inner and outer diameters, sq. ft.

A_B - Boiling area of the evaporator tube, Sq. ft.

B.T.P. - Bottom Tube Plate of the evaporator tube.

C - Specific heat of liquid, B.Th.U./(lb)(^oF).

D - Diameter of tube, ft.

h - Film coefficient of the feed liquid considering the entire length of the evaporator tube, B.Th.U./(hr)(sq.ft.)(^oF).

h_B - Film coefficient of heat transfer in the boiling section (boiling film coefficient), B.Th.U./(hr)(ft^2)(^oF)

H_A - Film coefficient of heating water (annular film coefficient), B.Th.U./(hr)(sq.ft.)(^oF).

H_W - Tube wall coefficient, B.Th.U./(hr)(sq.ft.)(^oF).

H_S - Steam film coefficient of heat transfer, B.Th.U./(hr)(sq.ft.)(^oF).

K - Thermal conductivity, B.Th.U./(hr)(sq.ft)(^oF/ft.).

L - Thickness of wall, ft.

Q - Total quantity of heat flow, B.Th.U./hr.

Q_B - Total quantity of heat flow through the boiling section of the evaporator tube, B.Th.U./hr.

Q_{NB} - Total quantity of heat flow through the non-boiling section of the evaporator tube, B.Th.U./hr.

T - Temperature, $^{o}F.$

T_f - Temperature of feed liquid, $^{o}F.$

T.T.P.- Top Tube Plate of the Evaporator tube.

U - Overall heat transfer coefficient based on the entire length of the evaporator tube, B.Th.U./(hr)(sq.ft.)($^{o}F.$)

U_B - Overall heat transfer coefficient in the boiling section of the evaporator tube, B.Th.U./(hr)(ft^2)(^{o}F).

U_{NB} - Overall heat transfer coefficient in the non-boiling section of the evaporator, B.Th.U./(hr)(sq.ft)(^{o}F).

v - Specific volume of vapour, Cu.ft./lb.

V - Velocity, ft./sec.

W_E - Evaporation, lbs./hr.

W_f - Rate of feed, lbs./hr.

ΔT - Log mean overall temperature drop, $^{o}F.$

ΔT_B - Integrated overall temperature drop in the boiling section of the evaporator tube, $^{o}F.$

ΔT_{NB} - Integrated overall temperature drop in the non-boiling section of the evaporator tube, $^{o}F.$

ΔT_m - Integrated overall temperature drop based on the entire length of the evaporator tube, $^{o}F.$

μ - Viscosity of liquid, lb./(ft.)(sec).

ρ - Density of liquid, lbs./cu.ft.

σ - Surface tension of liquid, dynes/cm.

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